

Proceedings of:

The Twenty-sixth Annual Meeting of

The American Society for

Precision Engineering

November 13 to November 18, 2011
Denver Marriott City Center Hotel
Denver, Colorado

The American Society for Precision Engineering (ASPE) is a multidisciplinary professional and technical society concerned with research and development, design, manufacture and measurement of high accuracy components and systems. ASPE activities encompass relevant aspects of mechanical, electronic, optical and production engineering, physics, chemistry, and computer and materials science. Membership is open to anyone interested in any aspect of precision engineering.

Founded in 1986, ASPE provides a new focus for a diverse but important community. Other professional organizations have covered aspects of precision engineering, always as a sideline to their principal goals. ASPE is based on the core of generic concepts necessary to achieve precision in any application; independent of discipline, ASPE intends to be the focus for precision technology – and to represent all facets from research to application.

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Preface

This book comprises the proceedings of the 26th ASPE Annual Meeting. The contributions reflect the authors' opinions and are published as presented to ASPE, without change. Their inclusion in this publication does not necessarily constitute endorsement by the ASPE, or its editorial staff.

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Welcome Note

Welcome to Denver and the Twenty-sixth Annual Meeting of The American Society for Precision Engineering. The growth and success of the Society reflects the increasing importance of precision engineering in a wide variety of fields, from manufacturing to microelectronics to basic science. The Society serves as a focus for precision engineers across all of these fields. The Annual Meeting has evolved into a premier international forum for the exchange of ideas and presentation of research results relating to precision engineering, metrology, controls, and system integration. Precision engineers and scientists from private industry, government laboratories, and universities meet to learn about the latest developments and to exchange ideas about the future directions of these technologies.

The 2011 ASPE Annual Meeting continues the tradition of offering two full days of tutorials presented by some of the foremost precision engineers and scientists in the world. The technical sessions and poster presentations offer the latest research results in the areas of ultraprecision machining, metrology, controls, precision grinding, micro-positioning, material processing, design, precision transducers, surface profilometry, and error compensation. The commercial exhibits will provide attendees with the opportunity to view and discuss the latest precision engineering equipment, products, and services.

The conference organizing committee is proud to present the program for the Twenty-sixth Annual Meeting of the ASPE. We welcome your participation and your feedback on how to make future meetings even better.

2011 Organizing & Technical Program Committee

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2011 Annual Meeting

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Program

2011 ASPE Annual Meeting

Time	Sun., Nov. 13	Mon., Nov. 14	Tues., Nov. 15	Wed., Nov. 16	Thurs., Nov. 17
8:00	Registration		Registration and Early Coffee		
9:00	Tutorials	Tutorials	Technical Session I	Technical Session III	Technical Session VI
10:00			Break	Break	Break
11:00			Technical Session II	Technical Session IV	Technical Session VII
12:00					
1:00			Lunch and Committee Meetings	Awards Lunch and Business Meeting	Lunch and Open Forum
2:00	Tutorials	Tutorials	Commercial Session	Technical Session V	Technical Session VIII
3:00			Break	Break	
4:00			Poster Session	Poster Session	
5:00					
6:00			Hospitality Hr.	Hospitality Hr.	
		Welcome Reception			
7:00		Keynote Address	Dinner at the Denver Museum Of Nature and Science		
8:00					
9:00					



Exhibit Open Hours

Keynote Address

John Peters

Chevron Corporation

Monday, November 14, 7:00 p.m.

Session Chair: Bradley H. Jared, Sandia National Laboratories

Mr. John Peters has worked for Chevron for 29 years. His previous assignments include Drilling and Completions Manager Chevron Argentina, Drilling and Completions Manager Chevron Australia, and Drilling Engineering Team Leader Chevron Energy Technology. He has a BS degree in Petroleum Engineering from Texas A&M University.

Chevron Corporation is the second-largest integrated energy company in the United States, and one of the largest corporations worldwide based on market capitalization. This work performed under the auspices of the U.S. Department of Energy by Lawrence Livermore National Laboratory under Contract DE-AC52-07NA27344.

Precision Engineering Applied to Deepwater Drilling and Completion

Deepwater drilling and completion faces many technical, safety, environmental and political challenges today, and yet it represents approximately 50% of all new oil discoveries worldwide. Currently, global deepwater oil production exceeds 5 million barrels of oil equivalent per day and is only expected to increase with rising global demand and depletion of “easy” oil reserves. Some technical challenges faced in deepwater drilling and completion includes operating in environments with pressures exceeding 20,000 psi, tensile loading exceeding 1,500,000 pounds and a dynamic sea state. Overcoming these technical challenges requires precision engineering in the areas of sensor measurements, data transmission, strength of materials, high performance mechanical designs and others.

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Technical Exhibitors

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Carl Zeiss Industrial Metrology	Precision Engineering Center – N.C. State University
Benstone Instruments	Precision Environments, Inc.
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Lion Precision	Trust Automation
Machine Dynamics Research Lab. – Penn State University	Western Environmental Corporation
Makino	Zygo Corporation

Commercial Session

Tuesday, November 15, 1:30 - 3:00 p.m.

Session Co-Chairs: Don Martin (Lion Precision)
and Marcin B. Bauza (Carl Zeiss IMT)

The Commercial Session will take place early in the week on Tuesday afternoon. This is a special session where companies and participants are free to open discussions on a less-formal basis and to promote interaction on a variety of topics. In the first part of this session, company representatives are invited to make brief presentations (five minutes) on new products associated with precision engineering. This provides participants with the opportunity to receive timely information on new technologies that have been commercialized into products and services, as well as information on advances that can be expected in the near future.

Open Forum “Horrobilia”

Thursday, November 17, 12:00 - 1:30 p.m.

Session Co-Chairs: Byron R. Knapp (Professional Instruments, Inc.)
and Brian P. O'Connor (Aerotech, Inc.)

To quote Chris Evans, “Precision engineering is plagued with painful re-invention and repetition of past failures.” The Horrobilia Session is intended to draw attention to some of these past failures, and reduce the chances of their repetition. It also prepares young engineers to deal with similar problems, understand how long it may take, how much patience may be necessary to find the exact source of a problem, but how easy they may be to fix!

In the tradition of Dr. Erwin Loewen’s and James Bryan’s past Horrobilia Open Forums, we have planned a very special session this year featuring introductory presentations by Erwin G. Loewen (retired), James B. Bryan (Bryan Associates), Robert J. Hocken (University of North Carolina at Charlotte), and Alexander H. Slocum (Massachusetts Institute of Technology). These world-renowned experts will recount problems they have encountered with expensive and embarrassing consequences such as days, weeks or months of frustration, machine downtime, and failed theories. The solutions to these problems are shockingly simple to understand, but not obvious.

All of this is for the purpose of warming up the audience to tell their own Horrobilia stories. Audience members will be encouraged to participate by recounting their own problems that have cost more money, took more time to solve, and were even easier to fix.

Erwin Loewen



Erwin Loewen spent a long professional career in the field well before it acquired the name precision engineering. Working in both industry and academia he specialized in the production of diffraction gratings with tolerances in the nanometer domain over significant areas. This is well before the word became so famous. Problems that had to be overcome make fascinating stories, both technical and personal. He is co-author of one of the few texts on the technology and history of the field. He is a life member of ASME, Fellow of Society of Precision Engineers, Society of Manufacturing Engineers and the Optical Society of America. He has been a member of CIRP (International Society of Production Engineering research) serving

various terms in its metrology subcommittee. For 19 years he was chair of ASME standards committee B-89 on dimensional metrology. He also joined ASPE in 1987, and is an Honorary Member of the Society.

James B. Bryan



Jim Bryan retired in 1985 after 30 years as the Metrology Group Leader at the Lawrence Livermore National Laboratory, University of California.

Honorary Memberships: (1) The International Academy for Precision Engineering (CIRP), (2) The European Society for Precision Engineering. (3) The American Society for Precision Engineering. (4) The Society of Manufacturing Engineers.

Awards: (1) The SME 1977 International Research Medal. (2) CSPE 1979 Archimedes Engineering Achievement Award. (3) The ASME 1995 Dedicated Service Award. (4) The SME 2007 Masters of Manufacturing Award. (5) The ASME 2008 Eugene

Merchant Award. (6) The CIRP 2009 General Pierre Nicolau Award.

Patents: (1) The “Telescoping Magnetic Ball Bar” (for testing machine tool contouring accuracy), (2) The “Slow Tool Servo” (for diamond tool facing of asymmetric workpieces).



Robert J. Hocken

Dr. Hocken began his career at the NBS where he developed software correction of CMMs and the use of computer assisted theodolites (with Bill Haight) for large-scale stereotriangulation. He played a lead role in the development of the Automatic Manufacturing Research Facility, invented the laser tracker (with Kam Lau), and edited the first American measuring machine standard. In 1988 he came to UNC Charlotte as a chaired Professor. Here he built the internationally recognized Center for Precision Metrology. The Center performs research and educates students in manufacturing metrology. Dr. Hocken has continued to performed research in areas ranging from large-scale metrology to nanotechnology. Also he is active on the B89 committee for dimensional metrology. He is currently working with other universities on nanotechnology projects.



Alexander H. Slocum

Alexander H. Slocum is a member of the IEEE and a Fellow of the ASME. He received the S.B. (1982), S.M. (1983), and Ph.D.(1985) from MIT in Cambridge Massachusetts in Mechanical Engineering.

From 1983 to 1985 he also worked at the US National Bureau of Standards. He is currently the Pappalardo Professor of Mechanical Engineering at MIT. He is the author of seven dozen journal articles and 12 dozen conference papers. He is also the author of text/reference books Precision Machine Design (Dearborn, MI, SME 1985) and FUNdaMENTALS of Design (Cambridge, MA, MIT 2005, <http://web.mit.edu/2.75/resources/FUNdaMENTALS.html>).

Dr. Slocum was awarded a Department of Commerce Bronze Medal in 1995 for Federal Service, seven dozen patents issued/pending, and has helped create 11 products that have been awarded R&D 100 awards, each for annually being one of one hundred most technologically significant new products. Dr. Slocum received the Martin Luther King Jr. Leadership Award in 1999 and was the Massachusetts Professor of the Year in 2000. Dr. Slocum has also received the SME Frederick W. Taylor Research Medal, and the ASME Leonardo daVinci and Machine Design Awards. Dr. Slocum has completed numerous Ironman triathlon events, is a rescue certified SCUBA diver, an avid snowboarder, woodworker, and has for many years helped coach a FIRST robotics team with his wife Debra.

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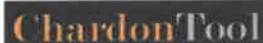
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2011 ASPE Lifetime Achievement Award



Graham J. Siddall

Graham Siddall received his BS degree in Production Engineering and Management with First Class honors from the University of Nottingham in England. He received his M.Sc. and Ph.D. degrees in Natural Philosophy (Physics) in 1971 and 1975, respectively, working under the guidance of Professor R.V.Jones at the University of Aberdeen in Scotland.

After working in surface metrology at Rank Taylor Hobson in Leicester, England, Dr. Siddall was awarded a Lindemann Research Fellowship to the United States where he joined the team at Stanford University working on the NASA/Stanford Gyro Relativity Experiment. His work at Stanford focused on developing the metrology techniques and equipment needed to manufacture the very precise spherical quartz rotors used in the cryogenic gyroscopes for the GP-B space mission. Following this, Dr. Siddall joined Hewlett-Packard

Labs in Palo Alto, working in the X-ray and E-beam Lithography group, primarily in the area of precision stage design. He went on to become the Vice President of research, development and engineering at GCA Corporation which marked the beginning of a long and distinguished stint in semiconductor capital equipment. Following GCA, in 1988 Dr. Siddall joined Tencor Instruments in California as Vice President of Technical Marketing and later became its first Chief Operating Officer. Over the next ten years the Company grew tenfold and merged with KLA in 1997, to form KLA-Tencor Corporation, where Dr. Siddall led the multi-billion dollar Wafer Inspection Group as Executive Vice President. In July 1999, he left KLA to join Credence Systems, a semiconductor test equipment company, as its President and Chief Executive Officer and then later, as Chairman of the Board, before retiring in October 2005.

Today, Dr. Siddall resides in Monterey, California, and is actively involved with several private companies. He serves as Chairman of the Board of DCG Systems, Inc., a provider of leading edge design diagnostics and failure analysis technologies to the semiconductor industry, with recent acquisitions in the areas of nano-positioning and thermal imaging. He is also a Director of ReVera Inc., a leading producer of next generation metrology equipment for monitoring the composition and thickness of critical layers deployed in semiconductor manufacturing processes. He is also Chairman of the Board of Nuforce, Inc, a fast-growing consumer electronics company and a Member of the Advisory Board at Dali Wireless (Beijing) Co., Ltd.

Graham is a past president of the American Society for Precision Engineering.

Technical and Poster Sessions Index

The technical program of the 2011 ASPE Annual Meeting contains 150 papers on precision engineering advances. Papers that lent themselves to significant verbal interaction, concise but important discoveries, or strongly visual or tactile subjects have been selected for the poster presentation. Authors will be in attendance to discuss their work during the Poster Session on Tuesday, November 15, and Wednesday, November 16, from 3:30 to 5:00 p.m.

Session I

Interferometry and Optics

Tuesday, November 15, 2011, 8:30 AM - 10:00 AM

Session Chair: Robert D. Grejda (Corning Tropel Corporation) and Robert E. Parks (University of Arizona)

- 1. A Novel Distance-measuring Interferometer for Rotary-linear Stage Metrology with Shape Error Removal**
Di Regolo, J. A.; Ummethala, U. (KLA-Tencor) 15
- 2. Towards Fiber-coupled Displacement Measuring Interferometers**
Ellis, J. D. (University of Rochester); and Meskers, A. J. H.; Spronck, J. W.; Munnig Schmidt; R. H. (Delft University of Technology). 19
- 3. Evaluation of the Measurement Performance of a Coherence Scanning Microscope Using Roughness Specimens**
Badami, V. G.; Liesener, J.; de Groot, P. J. (Zygo Corporation) and Evans, C. J. (University of North Carolina at Charlotte). 23
- 4. Fiber-optic Interferometry for X-ray Microscopy**
Nazaretski, E.; Eom, D.; Kim, J.; Yan, H.; Chu, Y.S. (Brookhaven National Laboratory); Shu, D. (Argonne National Laboratory); and Braun, P-F. (attocube systems AG) 27
- 5. Research in Novel Polishing Processes for Off-axis Mirror Segments**
Walker, D. D.; Yu, G. (Glyndwr University); Evans, R. (OpTIC-Glyndwr); and Li, H. (University College London). 31

Session II

Precision Manufacturing

Tuesday, November 15, 2011, 10:30 AM - 12:00 PM

Session Chair: Brian P. O'Connor (Aerotech, Inc.) and Hans-Jochen Trost (MicroFab Technologies, Inc.)

- 1. Application of Precision Engineering to the James Webb Space Telescope Primary Mirror**
Lewis, J. (Ball Aerospace & Technologies Corp.) 35
- 2. Alignment Fixture for Kinematic Mounts of Scientific Instruments on Hubble Space Telescope**
Knapp, B. R.; Liebers, M. J.; Sanner, S. L.; (Professional Instruments Company, Inc.) 39
- 3. Nickel Plated Metal Mirrors for Advanced Applications**
Gebhardt, A.; Scheiding, S.; Kinast, J.; Risse, S.; Duparré, A.; Trost, M. (Fraunhofer Institute for Applied Optics and Precision Engineering IOF); Rohloff, R.-R.; Schönherr, V. (Max-Planck-Institut für Astronomie MPIA); and Giggel, V.; H. Löscher (Carl Zeiss Jena GmbH) 43

4. Precision Engineering in Concentrating Solar Systems Shelef, B.; Erez, S. (Heliocentric Solar Technologies)	47
5. Continuous Contact Control for Microcontact Printing Using Precision Position Stage with Optical Feedback Petrzelka, J. E.; Hardt, D. E. (Massachusetts Institute of Technology)	51

Session III

Multi-Axis Machine Design

Wednesday, November 16, 2011, 8:30 AM - 10:00 AM

Session Chair: Byron R. Knapp (Professional Instruments Company, Inc.) and Mark Kosmowski (Electro Scientific Industries, Inc.)

1. Progress on High Precision Maglev Stage Technology for Parallel High-Throughput Reflective Electron Beam Direct Write Lithography Ummethala, U.; Clyne, J.; Di Regolo, J.; Hale, L. C.; Hench, J.; Wei, Z. (KLA-Tencor Corporation) and Angelis, G. Z.; van Lievenooogen, A. (Philips Innovation Services)	55
2. Design and Experimental Characterization of a Large Range XYZ Parallel Kinematic Flexure Mechanism Awtar, S.; Ustick, J. E. (University of Michigan - Ann Arbor)	58
3. Design and Characterization of a Visual Metrology Instrument with a 5-axis Motion System Lawson, B. L.; Montesanti, R. C.; Edson, S. L.; Horner, J. B.; Witte, M. C.; Di Nicola, P.; Kalantar, D. H.; Wallace, R. J.; Hughes, J. D.; Antipa, N. A. (Lawrence Livermore National Laboratory); Heidl, B. F.; Vonnhof, S. A.; Howland, F. A. (Akima Infrastructure Services, Inc.); and Ludwick, S. J. (Aerotech, Inc.)	62
4. Design and Testing of a High Accuracy Robotic Single-cell Manipulator Yoon, J. Y.; Trumper, D. L. (Massachusetts Institute of Technology)	64
5. Development of a 3D Surface Mapping System to Inspect Capsule Fill-tube Assemblies used in Laser-driven Fusion Targets Buice, E. S.; Antipa, N. A.; Bhandarkar, S.; Biesiada, T. A.; Conder, A. D.; Dzenitis, E. G.; Flegel, M. S.; Hamza, A. V.; Heinbockel, C. L.; Horner, J. B.; Johnson, M. A.; Kegelmeyer, L. M.; Meyer, J. S.; Maranville, B. L.; Montesanti, R. C.; Reynolds, J.L.; Taylor, J. S.; Wegner, P. J. (Lawrence Livermore National Laboratory); and Alger, E. T.(General Atomics).	68

Session IV

Precision Metrology

Wednesday, November 16, 2011, 10:30 AM - 12:00 PM

Session Chair: Jonathan E. Ellis (University of Rochester) and Marcin B. Bauza (Carl Zeiss, IMT)

1. Measurement Comparisons Between Optical and Mechanical Edges for a Silicon Micromachined Dimensional Calibration Standard Tran, H. D. (Sandia National Laboratories); Emtman, C. E.; Zwilling, A. M. (Micro Encoder, Inc.); and Salsbury, J. G.; Wright, W. (Mitutoyo America Corporation)	71
2. A Six-degree-of-freedom Motion Characterization of Ballscrew-driven Stage Using a Single Unit of an Optical Encoder Lee, C.B.; Kim, G.-H.; Lee, D.-J.; Lee, S.-K. (Gwangju Institute of Science & Technology (GIST))	75
3. Verifying the Capability of a Three Axis Vibrating Micro-scale CMM Probe to Operate in a Noncontact Mode Claverley, J. D.; Leach, R. K. (National Physical Laboratory)	77

4. Ultra-precision Linear Motion Metrology of a Commercially Available Linear Translation Stage Fesperman Jr., R. R.; Donmez, M. A.; Moylan, S. P. (National Institute of Standards and Technology)	81
5. A New Way of Analyzing Motion System Repeatability for Nanometer Precision: 6-dof Point Repeatability Brown, N. (ALIO Industries)	85

Session V

Nano-Positioning

Wednesday, November 16, 2011, 1:30 PM - 3:00 PM

Session Chair: Pradeep K. Subrahmanyam (Data Physics Corporation) and Senajith B. Rekawa (Lawrence Berkeley National Laboratory)

1. A Two Degree-of-freedom SOI-MEMS Translation Stage with Closed Loop Positioning Koo, B.; Ferreira, P. M.; (University of Illinois at Urbana-Champaign); and Dong, J. (North Carolina State University)	87
2. Compliant Microgripper with Parallel Straight-line Jaw Trajectory for Nanostructure Manipulation Beroz, J. D.; Awatar, S.; Bedewy, M.; Hart, A. J. (University of Michigan)	90
3. Interferometric Controlled Planar Nanopositioning System with 100 mm Circular Travel Range Hesse, S.; Schäffel, C.; Katzschnmann, M. (Institut für Mikroelektronik- und Mechatronik Systeme GmbH); and Buechner, H.-J. (Technische Universität Ilmenau)	94
4. Non-Lithographically-Based Microfabrication of Precision MEMS Nanopositioning Systems Panas, R. M.; Cullinan, M. A.; Culpepper, M. L. (Massachusetts Institute of Technology)	98
5. Flexure-based, Parallel Nano-positioning Robot Van Doren, M. J. (Van Doren Designs, LLC); Dornfeld, W. H. (ASML Wilton); and Marsh, E. R. (The Pennsylvania State University)	102

Session VI

Precision Machining Processes

Thursday, November 17, 2011, 8:30 AM - 10:00 AM

Session Chair: Mark A. Stocker (Cranfield Precision) and Allen Y. Yi (The Ohio State University)

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Technical Papers

A NOVEL DISTANCE-MEASURING INTERFEROMETER FOR ROTARY-LINEAR STAGE METROLOGY WITH SHAPE ERROR REMOVAL

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1. INTRODUCTION

A novel distance-measuring interferometer (DMI) with 10 axes of measurement has been developed for Reflective Electron Beam Lithography (REBL), a new technology being developed at KLA-Tencor under DARPA contract HR0011-07-9-0007. The REBL architecture features a continuously rotating platter of wafers which also slowly translates under a digital e-beam projector similar in concept to a digital light projector (DLP). Within a small ($100\text{ }\mu\text{m} \times 6\text{ }\mu\text{m}$) stationary field, it projects a dynamic, digital pattern that exposes patterns that are stationary on the moving wafers. The motion of the rotary-linear stage must be measured in six degrees of freedom in real-time to provide both feedback for controlling stage motion, including the magnetic spindle bearings, and feed forward for steering the e-beam. Ultimately the planar motion of each wafer must be tracked to the nanometer level with a portion allocated to stage metrology and error separation.

Within the confines of the REBL vacuum chamber, the 10-axis DMI system occupies a remarkably compact space and provides the following measurement functions (see figure #1):

- With respect to the chamber, three plane-mirror interferometers (X1, X2 and Y) combine to measure the X-Y- θ_z motion of the X-axis stage, which has conventional cross-roller bearings and is driven with a pair of linear motors.
- Operating in the same horizontal plane as the previous ones and with respect to the X-axis stage, three interferometers (R1, R2 and R3) fitted with cylindrical lenses measure the radial motion of a cylindrical mirror on the rotary stage. The redundant radial measurement allows the out of roundness of the mirror to be identified and removed from the measurements.
- These six interferometers in combination with an angular encoder measure the X-Y- θ_z

motion of the rotating and translating wafer platter with respect to the chamber.

- The remaining four interferometers (Z1, Z2, Z3 and Z4) measure differentially in the axial direction between an annular plane mirror on the rotary stage and individual stick mirrors on the chamber. Although the interferometer optics travel with the X stage, they combine to measure the θ_x - θ_y -Z motion of the rotating and translating wafer platter with respect to the chamber. As before, the redundant axial measurement allows the circular non-flatness of the mirror to be identified and removed from the measurements.

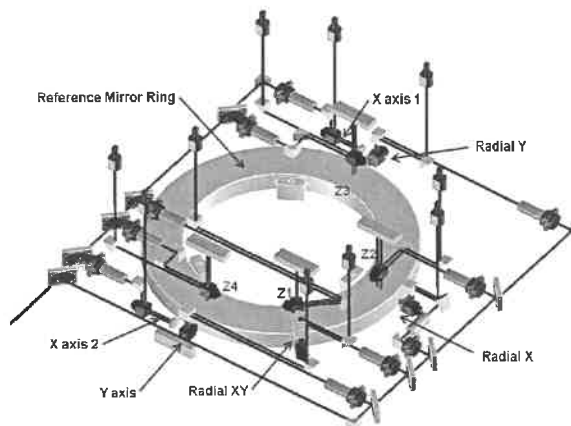


Figure 1

This paper presents the design of the 10-axis DMI system and techniques used to align the optics, and measured performance of the integrated system. Another paper in this proceeding (Hale, Ummethala and Hench) describes the details of the shape error removal using the multi-probe method.

2. SYSTEM DESIGN

The DMI metrology system provides a high resolution measurement system that accurately measures the movement of the rotating platter

relative to the metrology backplane (chamber lid & column).

The REBL DMI metrology system employs plane mirror interferometers that are highly sensitive to the angular misalignment of the input beam. Conversely this type of interferometer is relatively insensitive to linear displacement of the input beam. For this reason it is important that the mounting interface scheme minimize any angular deflection of the optical components as vacuum is applied to the chamber.

When mounting an Interferometer based metrology system in a vacuum chamber, it is important to consider the deflection of the mounting surfaces as large atmospheric forces interact with the outer surface of the chamber.

Depending on the mounting configuration of the metrology system and the physical shape of the chamber, both linear and angular deformation of the mounting surface can occur. This deformation can adversely affect the alignment and performance of the system. To control any adverse effects, the mounting interface must be carefully designed.

In order to mitigate the effects of chamber lid deflection, the REBL system uses four gimbal flexures. These flexures add compliance between the lid mounting surface and the metrology backplane. The stiffness of these flexures are designed to provide maximum modal stiffness without deforming the base of the metrology backplane. Figure 2 shows the four flexures mounted to the back of the metrology backplane along with four Z reference mirror mounts. The side facing down will be populated with beam steering optics.

Finite element analysis is used to determine a mounting location of the flexures. The flexure positions are arranged such that they deflect equally to provide a rectilinear motion as vacuum is applied to the chamber with a minimal angular change.

The flexures are attached to the lid by spherical rings which are clamped by collets mounted to the chamber lid after the relative alignments have been established with auxiliary tooling. (See figure # 3) The collets are designed to provide a radial gripping force to the spherical rings bolted to the top of the gimbal flexures.

The line contact that is created between the spherical ring and collet provides a rigid connection between the lid and the metrology backplane while minimizing the thermal conductivity path from the chamber to the metrology backplane.

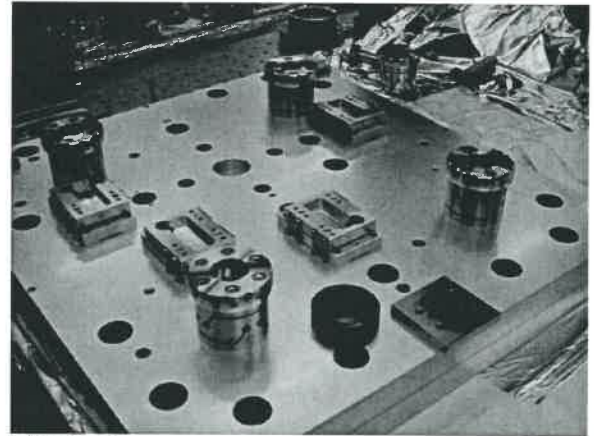


Figure 2

Unlike conventional collets like those found in lathes, the collets do not translate as they are tightened. This is accomplished by moving a tapered ring axially along the diameter of the collet flexure. This ring applies force to the outer diameter of the collet flexure via a tapered feature on the outer diameter of the collet. The collet is rigidly bolted to a mounting plate. The assembly is then bolted to a counter bored feature in the vacuum chamber. Clearance is provided in the mounting plate to allow the assembly to float in the X and Y plane. (See figure # 4)

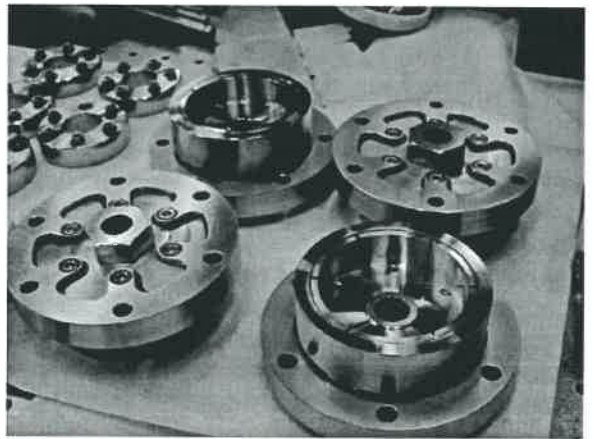


Figure 3

The combination of the spherical ring and collet provide a rigid attachment between the metrology backplane and the chamber lid with

the advantage of high modal stiffness, low thermal conductivity and minimal parasitic forces when secured after alignment.

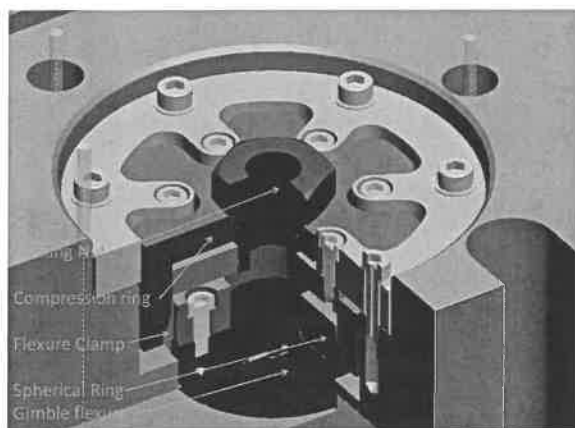


Figure 4

3. Metrology Alignment Strategy

Due to the space limitation and location of the metrology system in situ alignment is not possible once the system is under vacuum.

For this reason a metrology alignment strategy was developed based on the following criteria:

- Perform all 10 axis DMI alignments on bench Test Fixture
- Design a test fixture to simulate the final mounting configuration
- Transfer the aligned system to the final configuration by means of tooling that maintains system alignment.
- Provide global alignment of backplane optics to platter spin axis, X stage and Y axis.
- Test the system in atmosphere with access to all adjustments prior to full vacuum integration.
- Simulate the rotary stage to align radial DMI channels

To implement this strategy the following tooling and design features were developed:

- A tooling bridge that supports the full metrology system for atmospheric integration and testing.

- Tooling that accurately mates the stage metrology assembly to the distribution optics backplane
- Alignment tooling with quick release (magnetic bases and captive screws)
- A spherical joint to align the metrology system to maglev rotary axis
- A rotary reference mirror tooling stage for radial DMI alignment.

Figure 5 shows the metrology system mounted to its tooling (red anodized) frame. The frame doubles as a rigid support for shipping the metrology system and the atmospheric support of the system prior to full vacuum integration.

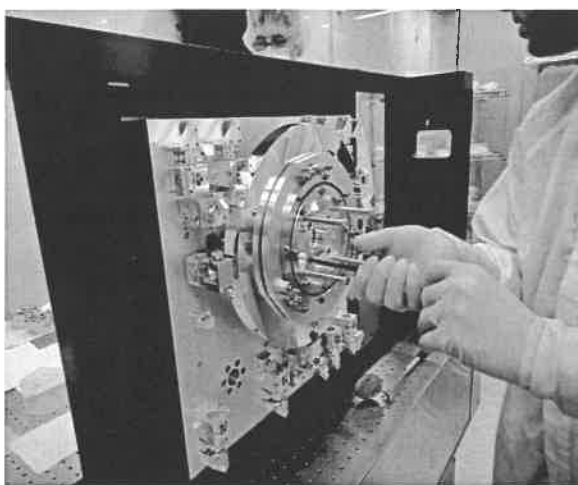


Figure 5

4. DMI Fine Alignment Effecting Shape Error Recovery

In order for the shape error recovery to be effective, the position of the interferometers placed at the three radial positions around the ring mirror and the additional four Z interferometers that measure the annular surface on the ring mirror must be carefully aligned to the correct angular position. The exact locations for these interferometers have been chosen to provide good shape recovery for both the cylindrical and annular mirror surfaces of the ring.

It is also critically important that the measurement beams from each interferometer track precisely the same trajectory around the reference ring mirror surfaces. The double pass interferometers used in the REBL system

produce two measurement beams. These beams can have positional differences from one interferometer to another depending on the alignment of the incoming laser. These differences include, beam separation, beam rotation and beam translational variations.

In order to minimize the effect of tracking error, the DMI's are aligned to a target placed on the ring mirror. Each interferometer is then aligned to the target via precision adjustments in the IFM mounts. This alignment insures that the measurement beams track along the same path around the mirror. The target provides a reference for both the annular mirror surface and the cylindrical surface so that the DMI's are aligned spatially to the same angular position. The target can be a shaped aperture or a position sensing device like a quad cell detector.

Conclusion:

When mounting a metrology system to a vacuum chamber, it is important to consider the mechanical deflection of the mounting interfaces and what impact they may have on the metrology system that is attached.

Understanding of the positional sensitivities of the optical components and the metrology subsystems will help in designing a mounting strategy that will have minimal impact on system performance and alignment.

Careful planning of the integration and alignment tooling strategy will help to ensure smooth system integration.

When applying redundant sensors to correct for shape errors of reference surfaces it is important that the sensors follow the same track so that the corrections are applied to a common spatial artifact.

Acknowledgements:

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TOWARDS FIBER-COUPLED DISPLACEMENT MEASURING INTERFEROMETERS

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INTRODUCTION

Heterodyne displacement measuring interferometry has been widely used to calibrate other measurement devices such as capacitive sensors, inductive sensors, and optical encoders because of its high dynamic range, high signal-to-noise ratio, and direct traceability to the length standards. The three main error sources that limit the performance of a displacement interferometry system are the laser frequency stability [1], refractive index fluctuations in non-common optical paths [2], and periodic nonlinearity in the measured phase due to source mixing, manufacturing tolerances, and imperfect alignment [3, 4, 5]. While the effects of the laser frequency stability and refractive index fluctuations can be mitigated by employing a highly stable reference laser and by performing the measurements in a well controlled environment or even vacuum, the periodic nonlinearity is difficult to eliminate.

Moreover, for practical interferometry systems, modularity is desired including fiber delivery to decouple the laser source (essentially a large heat source) from the interferometer. Fiber delivery further complicates periodic nonlinearity errors because changes in fiber stress and length can cause additional mixing which is difficult to assess and is time varying [6]. Additionally, fiber length changes can manifest a falsely measured Doppler shift signals, causing errors in the measurement.

Several periodic error free interferometer configurations share a common theme; the two optical beams with slightly different frequencies are spatially separated until the final interfering element [7, 8, 9]. In previous research, Joo-type interferometers, which use spatially separated input beams, have shown periodic errors down to a noise floor of 20 pm in the amplitude spectrum. However, two previously reported configurations, based on retroreflector and plane mirror targets, shown in Figure 1, are not ideal for industrial interferometry applications. Using a retroreflector as the measurement target limits the target motion to a single linear axis. Conversely, the

plane mirror target allows multi-axis motions but requires a large target mirror footprint.

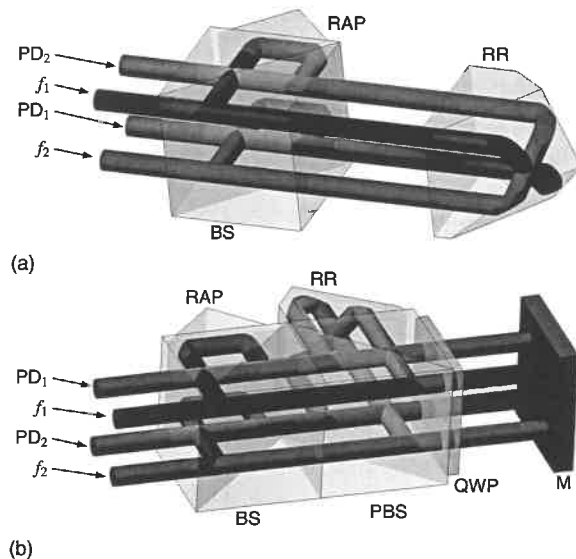


FIGURE 1. Solid models of the retroreflector (a) and plane mirror (b) Joo-type interferometers. The red and blue beams have slightly different optical frequencies.

In this proceedings, we present a general Joo-type interferometer with target mirror footprint comparable to traditional plane mirror interferometers. Additionally, we present a comparison between free space delivered and fiber delivered Joo-type interferometers.

The generalized Joo-type interferometer configuration (Joo-Gen), shown in Figure 2, has several advantageous features which can be leveraged to reduce these effects. Simple polarization cleanup on the input beams can limit the mixing due to the fibers. Also, an optical reference is generated within the interferometer block, which will eliminate fiber-induced Doppler shifts and can reduce periodic nonlinearity. This reference also gives a well defined thermal datum for the interferometer.

The Joo-Gen interferometer uses two spatially sep-

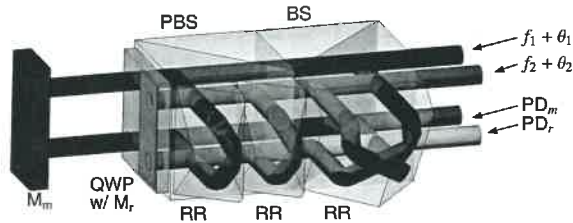


FIGURE 2. Generalized Joo interferometer. The red and blue beams have slightly different optical frequencies. The mirror on the quarter wave plate (QWP) serves as the reference mirror for eliminating fiber Doppler shifts. (PBS, polarizing beamsplitter; BS, beamsplitter; RR, retroreflector; M, Mirror; PD, photodetectors)

arated input beams with a known frequency difference as the optical source. The two beams enter and 50% is reflected at the BS and both beams travel to a large RR. When the beams reflect from the RR, they diagonally cross due to the RR's point symmetry, where they then travel back to the BS to interfere with their respective measurement arms.

The initially transmitted beams from the BS also transmit through a PBS and a QWP. One beam travels out to the target mirror and back while the other reflects off a reference mirror attached to the backside of the QWP. Both beams pass back through the QWP, changing their polarization state by 90° due to the double pass. Each beam then reflects off a smaller RR, oriented to displace the beams beneath their incoming beams. The two beams then reflect off the PBS and pass through the QWP again, on the way to traveling to their respective targets. The beams reflect back from their target, passing through the QWP again, changing their polarization state another 90°. In the PBS, the beams transmit due to their polarization state and then travel to the BS where they interfere with their respective reference arm.

When fiber coupling is used as the input, the phases of the two input optical beams fluctuate based on changing applied stresses, vibrations, and thermal perturbations. If this phase change is not addressed, the false Doppler shifts will be misinterpreted as length changes. The signals detected by PD_r and PD_m, including fiber fluctuations, can be described as

$$I_m \propto \cos(2\pi f_1 t + \theta_1) \cos(2\pi f_2 t + \theta_2 + \theta_m) \quad (1)$$

$$I_r \propto \cos(2\pi f_2 t + \theta_2) \cos(2\pi f_1 t + \theta_1 + \theta_r), \quad (2)$$

where $I_{m,r}$ are the measurement and reference irradiances; $f_{1,2}$ are the two nominal optical frequen-

cies; $\theta_{1,2}$ are the two fiber-induced phase fluctuations for $f_{1,2}$, respectively; and $\theta_{m,r}$ the phase changes from target motions. These equations simplify to

$$I_m \propto \cos(2\pi f_s t + \theta_1 - \theta_2 - \theta_m) \quad (3)$$

$$I_r \propto \cos(2\pi f_s t + \theta_1 - \theta_2 + \theta_r), \quad (4)$$

where f_s is $(f_1 - f_2)$. When each signal is compared to a conventional optical reference with a constant frequency of f_s , both signals suffer from errors based on the relative fiber-induced Doppler shifts. However, the phase difference between the two detected signals is $(\theta_1 - \theta_2 - \theta_m) - (\theta_1 - \theta_2 + \theta_r) = (\theta_m + \theta_r)$. When the reference mirror is stationary, the measured phase difference is the phase change due to Doppler shifts of the target mirror, irrespective of the fiber-induced Doppler shifts. The phase is related to target displacement by $\theta_m = 2\pi n N z / \lambda$, where n is the refractive index of unequal paths, N is the interferometer fold constant, z is the distance to the target, and λ is the wavelength of light traveling to the target.

This principle for canceling fiber-induced Doppler shifts is also valid for the Joo-RR and Joo-PM interferometers. Figure 3 shows a photograph of the Joo-RR interferometer under test using a retroreflector attached to an air bearing stage. The left side of the figure contains the fiber-coupled Joo-RR interferometer. The blue fibers are the input optical beams using polarization maintaining fibers and the orange fibers are multimode detection fibers attached to fiber coupled detectors. The source on the right generates the non-mixed heterodyne signal from a stabilized laser using two acousto-optic modulators driven at 39 MHz and 41 MHz. Portions are sampled from each upshifted optical beam and interfered prior to the fiber coupled to generate an optical frequency at f_s (2 MHz).

One common method to measure periodic nonlinearity is to measure the displacement of a stage driven in a nominally linear fashion. The linear displacement means the measured length change can be easily detrended with a linear slope and offset or a low order polynomial fit to remove the overall motion leaving the residual error. When periodic nonlinearity is present, the signal has an error at the first and second fringe orders which is a cyclical error with one and two periods manifesting every λ/N nanometers. For the Joo-RR and Joo-Gen interferometers, the first order periodic nonlinearity occurs every $\lambda/4$ (158 nm) whereas the Joo-PMI has its first order periodic nonlinearity occur every $\lambda/8$ (79 nm).



FIGURE 3. Photograph of the Joo-RR interferometer setup from Figure 1a. The source fiber coupling is on the right half of the table and the interferometer and air bearing stage is on the left half. Two blue PM fibers are used as the inputs and two orange MM fibers are used for detection.

One disadvantage to this method is it is difficult to distinguish optical periodic errors from vibrations and stage periodic error in a single scan. Even with multiple scans, stage periodic error is difficult to eliminate. However, air bearing stages are typically controlled with a linear scale which often have pitches on the order of micrometers, which means the periodicity is likely to not overlap with the optical periodic nonlinearity from a 633 nm HeNe laser with a fold constant of N . The vibrations and other noises can be mitigated by scanning the stage at several known velocities and identifying low noise regimes. Once those are known, the stage velocity and sampling parameters can be adjusted to ensure the periodic error (if any) will fall within a low noise regime, distinguishing it from other vibration errors.

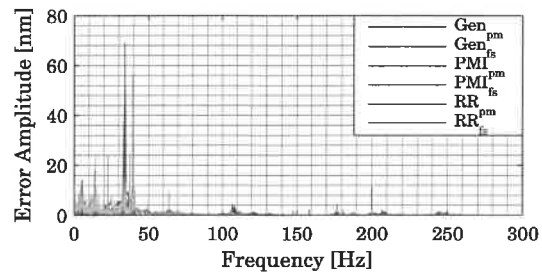
In this work, a stage was measured using each of the three interferometers (RR, PMI, & Gen) in both free space and fiber coupled configurations. The stage was driven in a trapezoidal fashion with a linear scan, dwell, and then linear scan at the same speed but opposite direction. The stage was measured with each interferometer, both free space and fiber coupled, at velocities of 100 nm/s, 1 μ m/s, 10 μ m/s, 100 μ m/s, 1 mm/s, and 1.6 mm/s, which was the maximum our data acquisition allowed while still acquiring enough points (without downsampling) to observe first and second order periodic nonlinearity.

Figure 4a shows the error amplitude spectrum in the low frequency regime measured by each interferometer with the stage traveling at 100 μ m/s. There is significant error below 50 Hz, as expected based on previous measurements in the laboratory environment and is due to acoustic coupling, air turbulence, and vibrations. The Fringe Order, FO , is the spatial

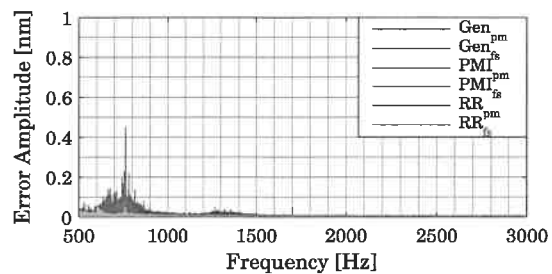
frequency normalized to the displacement in fringes or $\mathcal{F}\left(\frac{N\Delta z}{\lambda}\right)$, where Δz is the displacement and $\mathcal{F}$ indicates the frequency domain. A Fringe Order of 1 is one cycle every fringe of displacement, a Fringe Order of 2 is two cycles every fringe of displacement, and so on. For a stage traveling at a known velocity, v , a vibration at specific frequency f_{bw} manifests at a specific Fringe Order by

$$FO = \frac{\lambda f_{bw}}{Nv}. \quad (5)$$

The high vibration region in Figure 4a (<50 Hz) shows up 0.079 FO for the Joo-RR and Joo-Gen interferometers and 0.039 FO for the Joo-PMI, which is too slow to interfere with periodic nonlinearity assessment. At 100 μ m/s, the first and second FO appear at frequencies between 631 Hz and 2528 Hz, depending on the interferometer fold constant, which is a low noise regime as shown in Figure 4b



(a)



(b)

FIGURE 4. Error amplitude at a stage velocity of 100 μ m/s for the low frequency regime (a) and the high frequency regime (b) consistent with first and second order periodic nonlinearity. In (a), high vibration errors occur at frequencies below 50 Hz, however, the frequency regime where the periodic error is expected (b) has few vibrations. (subscript fs indicates free space, subscript pm indicates PM fiber coupled).

After converting the frequency axis to Fringe Orders, as shown in Figure 5, it is clear none of the interfer-

ometers in either free space or fiber coupled versions exhibit periodic nonlinearity error. The highest error appears to be in the fiber coupled Joo-Gen measurements but is still below 100 pm for the first order and below 50 pm for the second order.

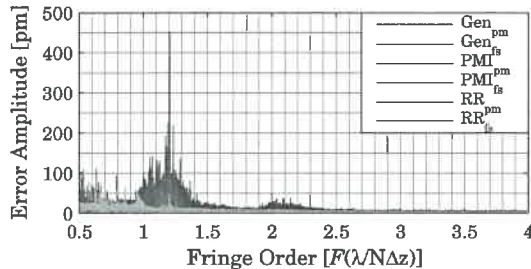


FIGURE 5. Measured periodic nonlinearity amplitude at a stage velocity of 100 $\mu\text{m/s}$. The error amplitude at the first and second fringe orders is very low, indicating almost no periodic nonlinearity.

Figure 6 shows the Joo-Gen periodic error measurements at various frequencies for the forward scan (solid) and backward scan (dashed). At a velocity of 1 $\mu\text{m/s}$, 1 FO, 2 FO, and 3 FO correspond to 6.3 Hz, 12.6 Hz, and 19 Hz, respectively, which is in the middle of the high noise regime (as shown in Figure 4a). However, as the stage velocity is changed, the peaks drop at those Fringe Orders, confirming those are vibrations and not periodic nonlinearity.

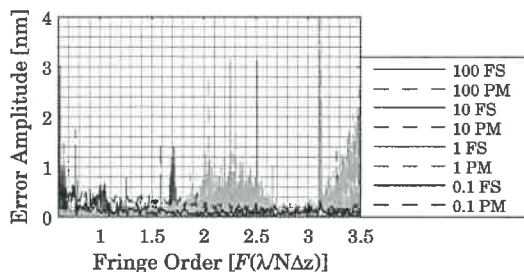


FIGURE 6. Measured periodic nonlinearity amplitude at four different stage velocities for both free space (FS) and PM fiber coupled (PM) Joo-Gen interferometers. The high error amplitudes at 1 $\mu\text{m/s}$ are due to an overlap in the low frequency, high vibration error regime shown in Figure 4a. (the number indicates stage velocity in $\mu\text{m/s}$.)

In summary, we present three different interferometer configurations with spatially separated inputs and assessed the periodic nonlinearity in both free space and PM fiber coupled variants. The periodic error was assessed by measuring a linear stage displacement and a Fourier domain analysis tech-

nique. Different stage velocities were used to distinguish vibration errors from periodic nonlinearity. Based on the results, each interferometer configuration showed no significant periodic nonlinearity (under 100 pm) in both free space and fiber coupled variants.

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Evaluation of the measurement performance of a coherence scanning microscope using roughness specimens

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INTERFEROMETERS FOR SURFACE ANALYSIS

Precision manufacturing has seen a steady increase in the use of high-speed, non-contact optical measurements of form and roughness. A dominant technology is coherence scanning interferometry (CSI), also known as scanning white light interferometry (SWLI), which provides quantitative maps of the surface topography for a million image points in a few seconds (Figure 1). CSI relies on interference fringe contrast, optionally combined with interference phase, to determine surface heights [1].

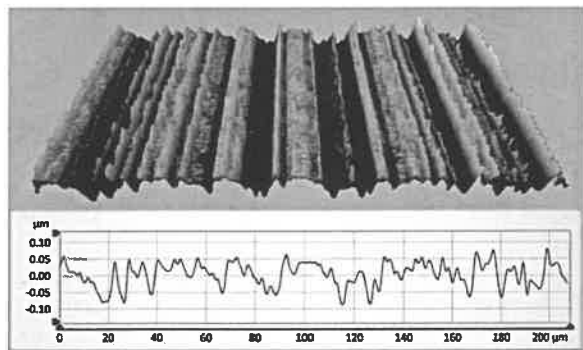


FIGURE 1. Areal surface topography map (above) and an extracted linear profile (below) of a Rubert 502 precision reference specimen as measured optically by CSI.

While recognizing the benefits of optical areal measurements, several groups have reported unexpected results from interference microscopes that compromise confidence in these tools [2]. In some cases, researchers have observed large differences in profile results depending on the data processing mode [3]. In other cases, average amplitude or R_a values measured with CSI have disagreed significantly with stylus measurements, in some cases by as much as a factor of two [4,5,6].

It has been shown that anomalous results for sub-micron R_a specimens can be overcome in many cases by switching to phase shifting interferometry (PSI) where feasible [3], or by using specialized CSI data analyses that make

use of phase information to improve measurement results on optically smooth parts [7]. However, the need to select between two or more measurement modes is a complication that has slowed the adoption of optical tools for routine measurements of surface texture.

Here we evaluate the performance of a new instrument (Figure 2) specifically designed to simplify the transition from stylus to optical measurements for metrology of precision machined, ground or polished surfaces.



FIGURE 2. ZeGage™ coherence scanning interferometer.

This instrument has a *single* measurement mode with a 1nm surface height repeatability midway in performance between traditional fringe contrast and phase measurement methods. Proprietary hardware and software obviate the height distortions sometimes observed in traditional CSI, significantly improving the measurement consistency and ease of use of the instrument.

The evaluation relies on a comparison of R_a values as determined by contact stylus and optical measurements. The results for fourteen commercial roughness specimens demonstrate good agreement over a wide range of surface textures from smooth to rough, with the best results obtained for magnifications that balance optical resolution with sufficient field of view to capture the dominant surface features.

REFERENCE SPECIMENS

Roughness measurement reference specimens are standardized in ISO 5436 Part 1:2000, where they are referred to as material measures. Types C (sinusoidal) and D (random) are appropriate for evaluating instrument performance and in applications where the user may wish to quantify a statistical parameter such as the average amplitude or R_a .

Rubert & Co Ltd manufactures a series of precision reference specimens made from electroformed nickel. They are most often used with stylus-type contacting surface measuring instruments as standardized in ISO 3274, but are increasingly being used for qualifying optical and non-contacting instruments. The present study concerns twelve of these standards as described in Table 1.

TABLE 1. Rubert roughness specimens and their nominal R_a values.

Specimen number	Pitch (μm)	Type	R_a (μm)
501	♦	Random	0.02
502	♦	Random	0.03
503	♦	Random	0.10
504	♦	Random	0.15
543	2.5	Sinusoidal	0.04
542	8	Sinusoidal	0.06
529	10	Sinusoidal	0.10
531	100	Sinusoidal	0.30
528	50	Sinusoidal	0.50
530	100	Sinusoidal	1.00
527	100	Sinusoidal	3.00
525	135	Sinusoidal	6.25

We also evaluated two HALLE Präzisions-Kalibriernormale random roughness specimens, one fine (KNT 2070/03 .024 micron R_a) and one coarse (KNT 2058/01 0.216 micron R_a), both calibrated at PTB. These high-quality calibration standards conform to ISO 5436.

METHODOLOGY

We performed stylus measurements on the Rubert specimens at the Center for Precision

Metrology at the University of North Carolina at Charlotte with a nominal stylus radius of $2\mu\text{m}$. We applied a Gaussian filter with a low-pass spatial period cutoff of $2.5\mu\text{m}$, per ISO 3274-1996. For stylus results on the Halle specimens, we rely on the certified R_a values from the manufacturer.

For the optical measurements, we employed Mirau objectives with 10X, 20X or 50X magnifications, according to the observed surface detail. The 1024 X 1024 camera provides field of view widths of $934\mu\text{m}$, $417\mu\text{m}$ and $167\mu\text{m}$, respectively. We applied an areal Gaussian filter with the $2.5\mu\text{m}$ -cutoff to the topography and calculated the R_a value from linear traces for direct comparison with the stylus results. For the random roughness specimens, the reported results represent an average of eight or more R_a measurements at different locations. There were no changes in the software configuration between the various samples.

RESULTS

TABLE 2. Measured R_a in μm for the Halle random roughness specimens.

	Stylus	Optical	
		10X	20X
Fine	0.0238	0.0237	0.0239
Coarse	0.2160	0.2370	0.2150

Table 2 compares the results for the Halle specimens, showing excellent agreement for the fine roughness specimen at both magnifications. The 10X measurement of the coarse specimen overestimates the R_a , possibly because of speckle noise from unresolved surface structures. The 20X magnification shows excellent agreement for both specimens.

Rubert random specimens 501, 502, 503 and 504 differ from the Halle specimens in that they contain more high-frequency roughness information; therefore, we chose 20X and 50X magnifications for the comparison. The results in Figure 3 and Table 3 again show good agreement between the stylus and optical R_a values. Figure 4 illustrates the agreement when using the 50X magnification compared to our $2\mu\text{m}$ radius stylus measurements.

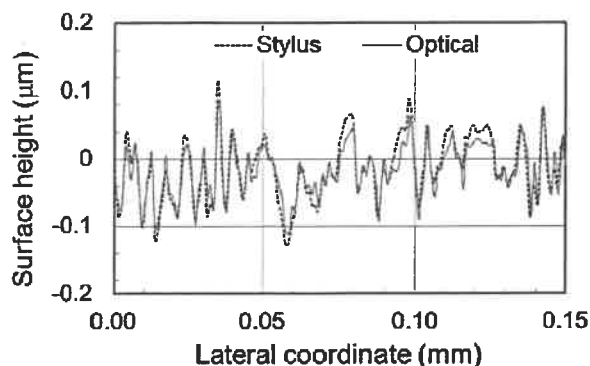


FIGURE 3. Comparison of stylus and optical profiles at 50X for the Rubert 502 specimen.

TABLE 3. Measured R_a in μm for the Rubert random roughness specimens.

	Stylus		Optical	
	2 μm	0.25 μm	50X	20X
501	0.015	0.019	0.012	0.016
502	0.035	0.038	0.031	0.036
503	0.067	0.099	0.076	0.095
504	0.119	0.150	0.130	0.150

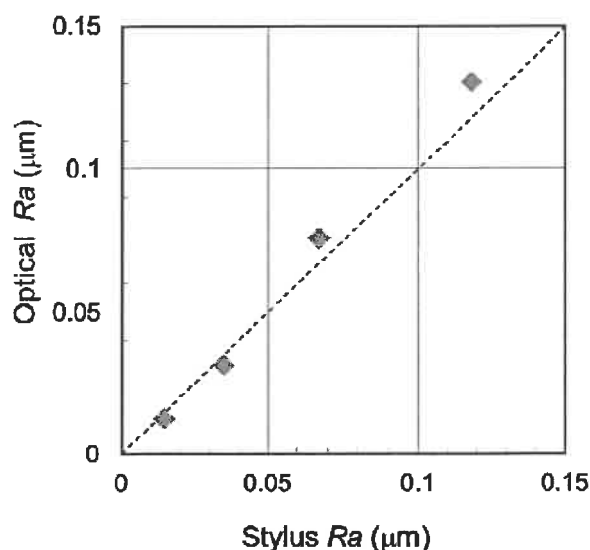


FIGURE 4. Graphical representation of the measurement results on Rubert random roughness specimens at 50X magnification. The diagonal line represents perfect correlation.

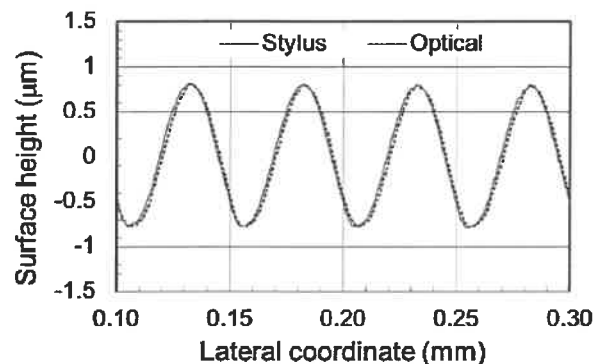


FIGURE 5. Comparison of stylus and optical profiles at 20X for the Rubert 528 specimen. A small phase shift has been added to more clearly separate the overlapping data plots.

TABLE 4. Measured R_a in μm for the Rubert sinusoidal specimens.

	Stylus	Optical	
	2 μm	50X	20X
543	0.018	0.021	-
542	0.060	0.051	0.048
529	0.097	0.090	0.108
531	0.315	0.315	0.316
528	0.507	0.515	0.506
530	1.009	1.050	1.012
527	2.995	3.066	3.025
525	6.389	6.362	-

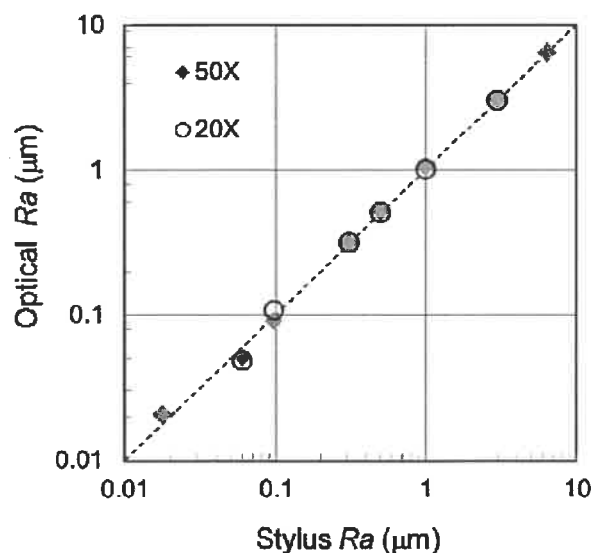


FIGURE 6. Graphical representation of the results on Rubert sinusoidal roughness specimens at 20X and 50X magnifications. The diagonal line represents perfect correlation.

Somewhat surprisingly, the 20X results are in better agreement with the stylus values provided by the manufacturer, which were obtained with a smaller, 0.25 μm radius tip. This is particularly evident for the 503 and 504 specimens. This phenomenon may relate to the sensitivity of CSI to optically unresolved surface roughness as described by Ettl et al. [8].

Figure 5 directly compares the stylus and optical profiles for the Rubert 528 specimen, and Table 4 summarizes the R_a values for all of the sinusoidal specimens. The 20X magnification has insufficient lateral resolution for the small-pitch 543 specimen and insufficient slope acceptance for the large-amplitude 525 specimen; while the 50X accommodates all eight specimens. The summary graph in Figure 6 plots the optical result as a function of the stylus measurement on a log-log scale, showing good agreement over a wide range of roughness amplitudes, for both the 20X and 50X magnifications.

CONCLUSIONS

As Paul Rubert has observed, "...many optical surface measuring instruments can be used in more than one mode of operation, each giving different results with certain surfaces, but with no obvious or a priori way for the operator to know which mode should be used in which case..." [9]. The evaluation here of a new coherence scanning interferometer shows progress towards overcoming this limitation. While the choice of objective does influence the result, the comparison of stylus and optical R_a measurements show good correlation across fourteen roughness specimens having highly varied surface texture using a single software mode.

We view the results as a significant step forward towards simplifying the desirable transition from traditional stylus instruments to high-speed, non-contact areal measurements for routine measurements of roughness and surface form in precision engineering.

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FIBER-OPTIC INTERFEROMETRY FOR X-RAY MICROSCOPY

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INTRODUCTION

The hard x-ray nanoprobe beamline (HXN) at the National Synchrotron Light Source II (NSLS-II) at the Brookhaven National Laboratory is aimed to achieve scanning fluorescence and diffraction capabilities with sub 10 nm spatial resolution and the ultimate goal of ~ 1 nm. Achieving sub 10 nm spatial resolution requires numerous developments of the x-ray focusing optics, novel design of the scanning microscope and implementation of state-of-the-art components. Due to stringent technical requirements for stability we consider the approach taken by the scanning probe microscopy community, e.g. development of smaller scale, stiffer mechanical components with the minimal generation of the heat due to mechanical motion and its encoding. One of the critical components of an x-ray microscope is a set of interferometers used to monitor the motion of translational/scanning stages and provide the feedback signal to compensate for possible thermal drifts in the system. Several commercial interferometers such as Renishaw [1], SIOS [2], Optodyne [3], and HP [4] provide viable options for x-ray microscopes. Having certain advantages such as capability of differential measurements, large working distances and low background noise, they also suffer from having large sensor heads, susceptibility to changes of ambient pressure/temperature (due to long optical path) and are not well characterized for long term stability of electronics. To achieve better performance needed for the HXN microscope we pursue the approach commonly used in the field of scanning probe microscopy for detection of small (micrometer scale) displacements using fiber-optic interferometers. The initial work on fiber-optic interferometry was performed by D. Rugar et al., and the authors have reported the

resolution of 0.1 Angstrom in a dc to 1 kHz bandwidth. [5] This work followed by numerous applications in different fields of scanning probe microscopy [6, 7] and culminated in the development of an interferometer capable of resolving 2 pm steps in a dc to 1 kHz bandwidth. [8] The main disadvantage of these interferometers pertains to a limited working distance (typically less than 100 μm). The reason for a limited working distance is the fact that laser light diverges as it exits the fiber and the intensity drops drastically on a scale of tens of microns away from a fiber end. The natural way to overcome this problem is to add a set of collimating lenses that will minimize divergence and will enhance the working distance, as schematically shown in Fig. 1. Aspheric lens installed inside of a collimating head allows to increase the working distance substantially.

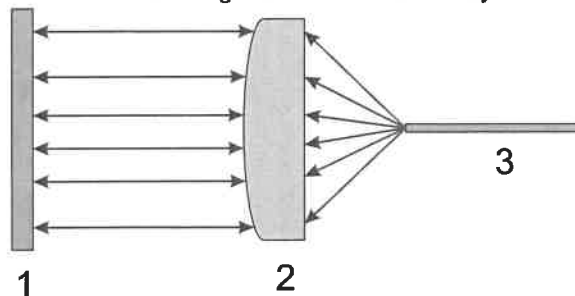


FIGURE 1. Schematic of a collimating head that uses an aspheric lens to increase the working distance. 1- reflector, 2 - aspheric lens, 3 - optical fiber

Working through an industrial partner (Attocube Systems AG) we have developed a multiaxis system (12 channels powered by a single laser source) that can be readily implemented in the x-ray scanning microscope. It has a large working distance (up to 100 mm) and the wavelength sta-

bilization (that includes both, temperature stabilization of the laser and referencing of the wavelength to the absorption line in a gas cell) that results in a noise floor of 250 pm and the long term stability of electronics which translates into a less than 1.25 nm drift over a 40 hours time period.

CHARACTERIZATION OF INTERFEROMETER

To verify performance of the interferometer we conducted three evaluation tests. The first experiment aimed to probe the long term stability of the interferometer hardware and electronics, the second one targeted measurements of the interferometer resolution and the third experiment determined requirements for alignment between the interferometer collimator head and the reflector.

Stability test

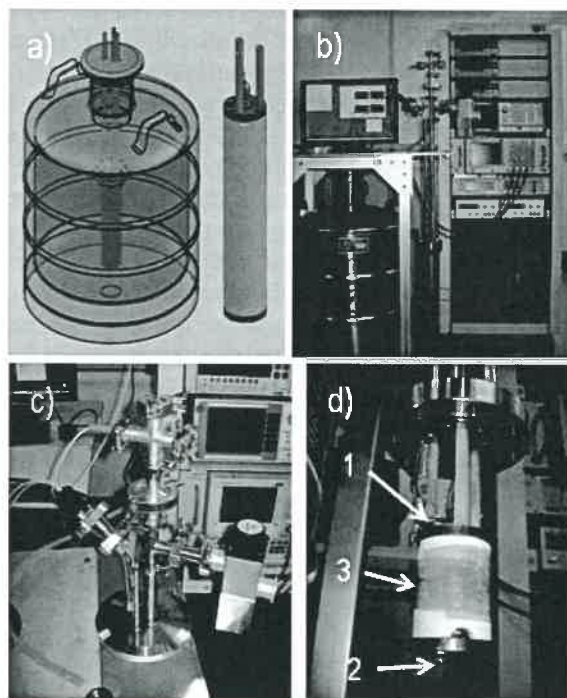


FIGURE 2. Cryogenic setup to test the stability of the interferometer. a) overall design of the cavity, vacuum chamber and the cryogenic dewar, b) overview of the setup with control electronics, c) view of the top flange d) test cavity. 1 and 2 indicate upper and lower flanges where the Silicon Diode thermometers are attached; 3 indicates the 50 mm length macor interferometric cavity.

To characterize the long term stability of the interferometer itself we constructed the cryogenic system shown in Fig. 2. The interferometric cavity

was manufactured from macor ceramics and has a length of 50 mm; during the experiment it was cooled to a liquid nitrogen temperature to reduce thermal expansion coefficient thus fixing the cavity length. The cavity was capped with two copper flanges, the reflector was mounted on one of the flanges and the interferometer head was attached to another one.

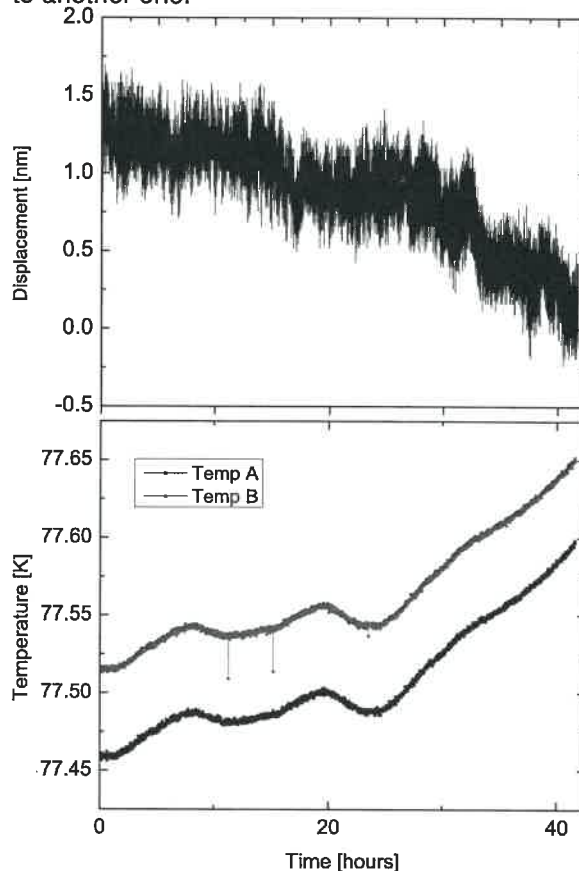


FIGURE 3. Long term stability measurement at liquid nitrogen temperature. Upper panel: interferometer readings as a function of time. Lower panel: temperature readings on the top and bottom flanges of the cavity, the observed drift of the distance measurements correlates with the temperature drift due to boiling off of liquid nitrogen.

Two calibrated Silicon Diode DT-670-CU-70L Lakeshore thermometers and Lakeshore 336 temperature controller were used to monitor the temperature of both flanges. The test cavity was placed inside stainless steel vacuum chamber and pumped out. All measurements were performed in vacuum and stability of the interferometer was determined by measurements of the drift as a function of time. Data acquisition was performed using Labview software specifically writ-

ten for these experiments. Fig. 3 depicts results of the measurements over 40 hours time period. The drift of approximately 1.25 nm was observed and the drift trend is consistent with temperature changes of the cavity due to boiling off of liquid nitrogen (see lower panel in Fig.3).

Resolution test

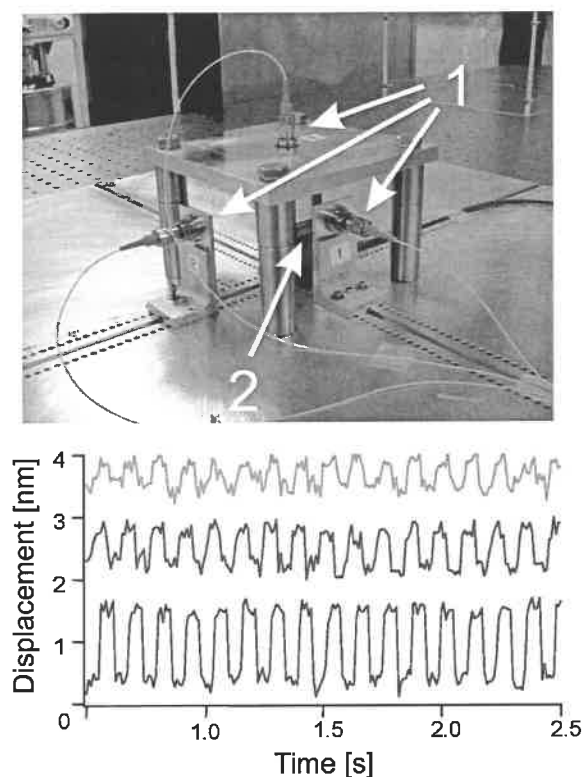


FIGURE 4. Setup for interferometer resolution test. Upper panel: photograph of the setup showing interferometer heads (1) and PI Picocube (2). Lower panel: 300 pm, 500 pm and 1000 pm steps performed by the Picocube in the z-direction and recorded by the fiber-optic interferometer. Interferometer head-reflector separation of set to 25 mm

To verify the noise floor and resolution of the fiber-optic interferometer we constructed a test bench shown in the upper panel in Fig.4. The fine steps were performed by a high performance scanner equipped with a built-in capacitive sensor (PI P-363 Picocube). The reflector was mounted on top of a scanner and interferometer heads were positioned around the reflector to monitor X, Y, and Z displacements. The distance between the reflector and the interferometer head could be adjusted in the range between 5 and 300 mm. Dur-

ing the measurements the setup was enclosed in a double wall insulating box to minimize temperature and pressure fluctuations. Sound absorbing foam attached to the walls of the enclosure reduced acoustic noise that may affect measurements. The spatial resolution measurements were performed at ambient conditions and 25 mm interferometer head-reflector separation; experimental results are shown in the lower panel in Fig.4. We performed 1000 pm, 500 pm, and 300 pm steps, the smallest steps can be clearly resolved.

Angular acceptance test

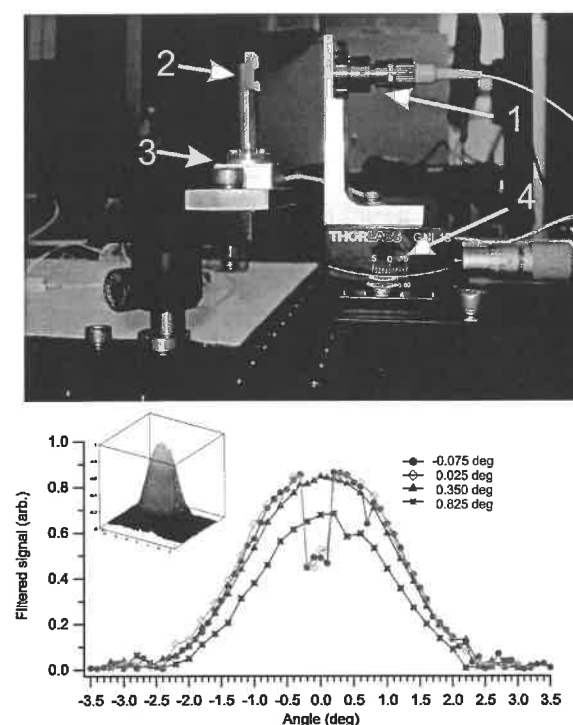


FIGURE 5. Setup to measure the angular acceptance of misalignment between the interferometer head and the reflecting surface. The cavity length is set to 20 mm, and key components are labeled as follows: 1 - interferometer head, 2 - reflector, 3 - Smaract SR-2013 rotary stage, 4 - goniometer stage. Lower panel: intensity of the interferometer signal as a function of angle defined by the Smaract rotary stage (3) for a given tilt angle set by the goniometer stage (4). Inset in the lower panel shows the 3D interferometer signal intensity plot as a function of rotation and tilt.

Angular acceptance for misalignment of the interferometer head with respect to reflecting surfaces is very important since in case of precise align-

ment requirements additional adjustment mechanism has to be implemented. For x-ray microscopy applications it is especially important since the position of x-ray optic must be monitored while its orientation angle may change over a small angle. Larger acceptance angle greatly simplifies measurement procedure and requirements. In our fiber-optic interferometer system the 1550 nm reflected laser light is focused on a 9 μm core of SMF-28 fiber and then transmitted to the interferometer controller. We performed an experiment to determine what misalignment is acceptable for operation of the fiber-optic system. The test setup is shown in the upper panel in Fig. 5. The interferometer head is mounted on top of a Thorlabs goniometer stage which provides tilt. The reflector is located 20 mm away from the collimating interferometer head and is placed on top of a Smaract SR-2013 rotary stage. We manually fixed the tilt and scanned the rotary stage while recording the intensity of the interferometer signal at the same time. Line scans for different tilt angles are shown in the lower panel in Fig. 5. In the inset we show the 3D interferometer signal intensity plot as a function of tilt and rotation. As seen in the Figure, the non-zero interferometer signal intensity is observed at rotation/tilt angles as large as $\pm 3.4^\circ$ making the alignment easy and avoiding any mechanism specifically built for alignment of interferometer heads.

CONCLUSION

In summary, working with the industrial partner we have developed a fiber-optic based interferometer suitable for the x-ray microscope to be built at the HXN beamline at NSLS-II. We performed thorough characterization of the system and determined the long term stability, ultimate resolution and angular acceptance for misalignment between the interferometer head and the reflecting surface. Compact design and superior characteristics of the interferometer will allow to monitor displacements and drifts of multiple components of the HXN microscope with sub-nanometer accuracy.

ACKNOWLEDGMENTS

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RESEARCH IN NEW POLISHING TECHNIQUES FOR OFF-AXIS MIRROR SEGMENTS

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INTRODUCTION

Monolithic primary mirrors for ground-based optical/near-IR telescopes have probably reached their limit with the 8.4m diameter mirrors produced by the Steward Mirror Lab of the University of Arizona. The primary mirrors of the 10m-class telescopes in service (Keck 1 and 2, etc.) are 'tiled' with hexagonal segments. This owes as much to the challenges of handling, transport of large pieces of glass to a remote mountain site, and the size of the vacuum-coating facility required, as to the innate difficulty of fabrication.

Future key astronomy programmes will be limited by photon-starvation, for example, the search for earth-like planets in the habitable zones around other stars, and stars and galaxies at the first epoch of their formation after the Big Bang. Since detectors are approaching the ultimate quantum efficiency, the way forward has to be larger collecting aperture. In response to this, the European Southern Observatory ('ESO') is advanced in its design of the European Extremely Large Telescope, ('E-ELT') [1]. Originally conceived with a 42m aperture f/1 primary, it has recently been re-scaled to 39.3m [on cost grounds].

The revised E-ELT primary mirror design comprises 798 hexagonal segments, each approximately 1.45m across-corners, 50mm thick, and with off-axis aspheric form. The total manufacturing requirement is for 931 segments, allowing one spare for each family of segments, given the six-fold symmetry of the assembled mirror [2].

Issuing a tender for one such mirror segment might elicit a response quoting a delivery time of a year or more. The manufacture of 931 such

segments in a few years, demands a new way of thinking about the problem of large optics production. In particular, the emphasis turns from repeating a *prototype* manufacturing cycle, to implementing something more akin to a *production line*. This requires investment in process-determinism, automation of fabrication, metrology and handling, minimization of manual interventions, and streamlining of data-flow for measurement analysis and process-control.

With this in view, process R&D has focused on deterministic ultra-precision grinding of the hexagonal segment by Cranfield University, and CNC polishing by the consortium based at OpTIC. The most challenging issue has proved to be as expected – the ability to control the edges of the segments, avoiding edge-roll.

THE FACILITY AT OPTIC IN NORTH WALES

Establishment of the Facility

In 2005 the segment opportunity was on the horizon, and this led to the formation of the National Facility for Ultra-Precision Surfaces, hosted by OpTIC-Glyndŵr, and located in the North Wales Electro-Optics Cluster of companies. At this stage, the principal equipment comprised the Zeeko IRP600 and 1200 CNC polishing machines, and the BoX Ultra-precision grinder [3], (since re-located back to its builder – Cranfield University). The Facility operates as a collaboration between OpTIC (now run by Glyndŵr University), University College London, Cranfield University and industrial partners including Zeeko Ltd.

Facility development for prototype segments

OpTIC-Glyndŵr has been awarded a contract by ESO to manufacture seven prototype mirror segments representing a cluster at the edge of

the original 42m mirror. This stimulated the establishment of a pilot segment fabrication plant. This is furnished with the first IRP1600 (1.6m capacity) Zeeko CNC polishing machine, full-aperture test-tower, other associated metrology capability, and bespoke segment handling equipment. This plant is supported by the IRP1200 and 600 Zeeko machines for R&D.

One of the dominant causes of poor process-convergence in CNC polishing has proved to be inadequate registration of the coordinate frames of the metrology data, the part, and the CNC tool-path for corrective-polishing. To manage this issue, all segment metrology during polishing cycles is conducted *in-situ* on the CNC polishing machine. This preserves the integrity of the coordinate-frame registration, as well as greatly reducing the handling of segments compared with off-machine metrology.

The key challenges for in-situ metrology are vibration and the thermal environment. Neither the test tower nor the IRP1600 machine (Figure 1) is vibration-isolated. Rather, we have adopted simultaneous phase interferometry from 4D Technology Corp., which effectively freezes vibration and enables effective data-averaging.



FIGURE 1. Master Spherical Segment on the Zeeko IRP1600 CNC polishing machine, under the full-aperture Optical Test Tower at OptIC-Glyndŵr

In the telescope, the segments must work together to bring light to a common focus, and so maintaining matching base-radius is critical. We have developed two methods to assert this:- i) differential measurement of the segments with respect to an over-size Master Spherical Segment (MSS), and ii) an independent test using an autocollimator-pentaprism non-contact optical profilometer (the 'NOM' – nanometer optical metrology'). The latter directly measures surface-slopes, which are numerically integrated to give a height-profile. This profilometer is designed for interchangeable use, either in the lab, or deployed on the IRP1600 (Figure 2).

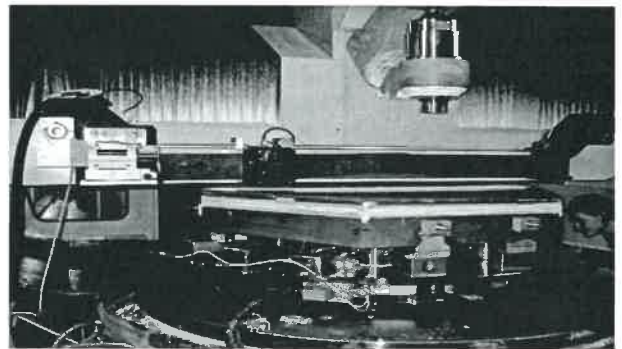


FIGURE 2. Testing the MSS using the NOM on the Zeeko IRP1600 CNC polishing machine

MASTER SPHERICAL SEGMENT (MSS)

The MSS was loose-abrasive ground at a sub-contractor to $R=84m$ concave. On the Zeeko machine at OptIC it was smoothed with rigid polishing tools of limited flexibility, and pre- and corrective polished under very conservative conditions: primarily an R80 bonnet delivering a 20mm spot-size.

The Zeeko process moves a small "spot of action" around the surface of the part in a pre-determined path. Modifying the traverse speed effectively changes local dwell-time and so provides capability to rectify measured form errors. The spot-of-action can be created by a standard Zeeko spinning, precessed bonnet covered with polishing cloth and operating in the presence of re-circulated polishing slurry. Alternatively, a wide range of bespoke tooling may be deployed, providing effective automation of more traditional polishing techniques.

The first challenge proved to be achieving the overall form whilst managing mid spatial frequency features. For the MSS, this required

both bonnet polishing, and smoothing using a hard tool of limited flexibility.

The MSS was qualified using the Optical Test Tower in its spherical measurement mode. Test results for the full optical aperture may be seen in Figures 3 and 4. Confirmatory tests and establishment of base radius were conducted on-machine using the NOM profilometer.

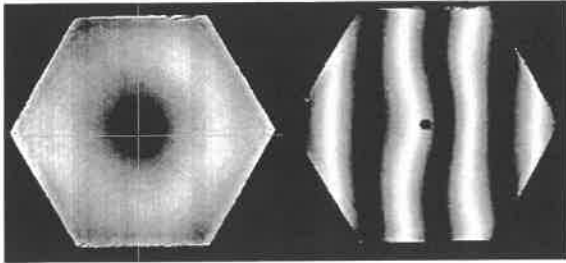


FIGURE 3. 1.5m a/c Master Spherical Segment phase map (left) and synthetic interferogram (right), showing 16.8nm RMS in optical area

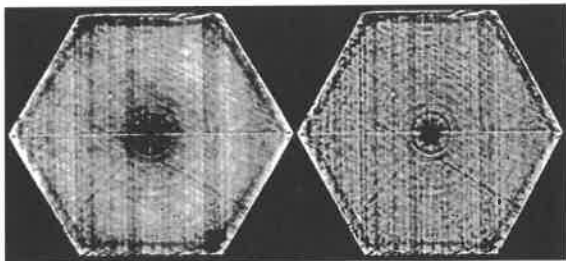


FIGURE 4. 1.5m a/c Master Spherical Segment filtered phase maps: 5.5nm RMS for spatial scales <250mm (left), and 4.2nm RMS for spatial scales <100mm (right)

The profilometer itself was validated using 'round robin' test parts that have been circulated through European laboratories with similar equipment. Discrepancies in the resulting profiles were typically ~ 40nm P-to-V, with the results from the OpTIC instrument near the mean. This was remarkably good, given that profilometers elsewhere typically operate under tight environmental conditions, in contrast to the on-machine environment at OpTIC.

THE FIRST ASSPHERIC SEGMENT

This Zerodur segment was delivered from Schott machined to the final hexagonal shape, and with rear mounting details. It was then ground to the off-axis aspheric form on the Cranfield 'BoX'

ultra-precision grinding machine [3], with a surface RMS form error of approximately 1 micron RMS and sub-surface damage < 10 microns. The grinding process left a couple of hundred nm of mid-spatial frequency features and some other artifacts.



FIGURE 5. First aspheric segment in process

The conservative polishing process deployed on the MSS has been adopted for the first aspheric segment. The effect of aspheric misfit of hard smoothing tools of limited flexibility was a concern, and was therefore validated on a sub-scale hexagonal witness part (pre-formed with a cylindrical term best-fit to the asphericity of the most severe segment). No significant mid-spatics were introduced. This smoothing process, as applied to the first full-size aspheric segment, has been shown to be effective in removing the BoX residuals, and corrective polishing is proceeding (Figure 5).

EDGES AND PROCESS-SPEED

Development of edge treatment is proceeding in parallel with early stage processing of the first segment [4]. With the near-Gaussian polishing spot from a precessed bonnet, edge down-turn can be avoided by under-sizing the boundary of the raster tool-path, leaving a turned-up edge. Our strategy is to start with large spots for high removal-rate (leaving a wide and high edge up-stand) then progressively move to smaller spots, reducing up-stand width and height. The part is finished with a smoothing tool.

Figure 6 shows a 200mm across corners witness part after pre-polishing with a ~50mm spot-size. Figures 7 and 8 show the same part after polishing with a ~20mm spot, leaving a turned-up edge. Figure 9 shows the edge definition after smoothing, followed by trimming of the bevel by 0.5mm. The triangular masks

visible in the figures are placed with the sharp points defining the precise start of the bevel (edge of polished area). The central mark is a fiducial applied to the witness part to track absolute depth of removal.

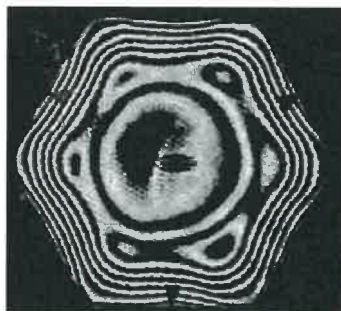


FIGURE 6. 200mm a/c hexagonal witness part, after pre-polishing with R160 bonnet and ~50mm spot size

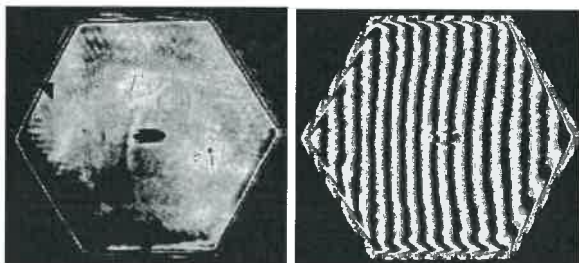


FIGURE 7. The part of Figure 6 after subsequent polishing with R80 bonnet and 20mm spot-size.

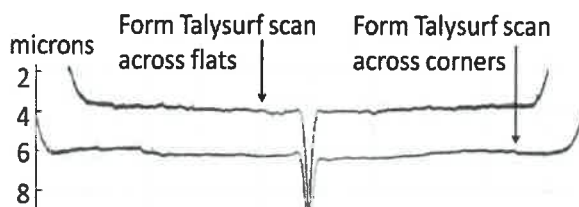


FIGURE 8. Form Talysurf profilometer scans for the part of Figure 7

A sharp down-turn at the extreme edge (Figure 9) was traced to the smoothing stage. Subsequent experiments show that this can be avoided, and the process-chain is currently being re-optimized in readiness for deployment on the second segment. In parallel, an R320 bonnet delivering a 100mm spot size will be brought into service, and this faster process chain prepared for the third segment.

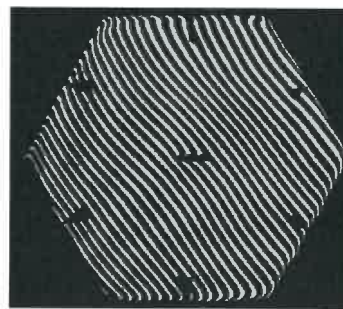


FIGURE 9. Edge definition of the part of Figure 7, after smoothing and 0.5mm of beveling.

CONCLUSION

The basic process chain and metrology are established, and quality demonstrated on the Master Spherical Segment. The process-chain is compatible with segment asphericity, and the first aspheric segment is proceeding as expected. In parallel, process-speed and edge-quality are being optimized in readiness for the second and subsequent segments.

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Application of Precision Engineering to the James Webb Space Telescope Primary Mirror

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observe red shifted light emitted during the formation of the first galaxies formed after the big bang.

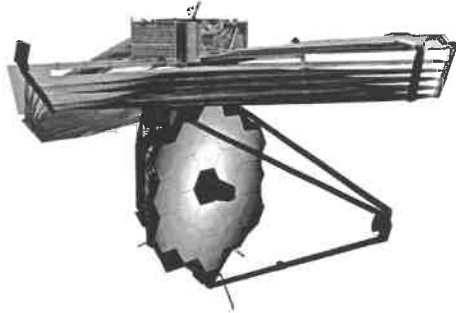


FIGURE 2. Artist rendition of JWST

OBSERVATORY ARCHITECTURE

The telescope is comprised of 18 primary mirror segments, a secondary mirror, tertiary mirror and fine steering mirror. The 18 primary mirror segments, shown in figure 3, are each supported by a hexapod to allow them to be aligned precisely in 6 degrees of freedom creating one large mirror surface. Additionally each primary mirror segment has a radius of curvature actuator that can minutely bend the mirror to match their radius of curvature to each other.

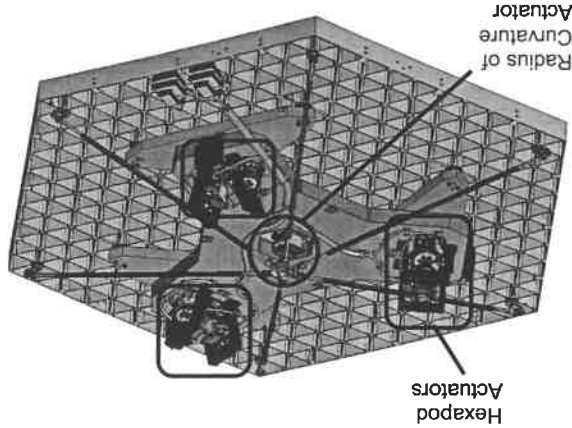


FIGURE 3. JWST Primary Mirror Segment.

The primary mirror segment is 1.5 m point to point and is made from beryllium lightweighted

INTRODUCTION

The 18 primary mirror segments for the James Webb Space Telescope (JWST) are 1.5 meter hexagonal beryllium optics with a surface error less than 25 nanometers rms. They include a hexapod positioning system and radius of curvature adjustment and operate at temperatures as low as 32K. To adjust the mirror a two stage linear actuator has been developed to provide 10 nanometer step size with 21 mm range using a single motor. The actuators cannot be present during polishing and the deformation caused by actuator installation must be characterized and the inverse polished into the mirror. This creates a requirement that the deformation due to actuator installation be extremely repeatable so that upon re-installation of the actuators the mirror has the correct surface figure.

JWST SCIENCE MISSION

JWST explores the gap between the distant microwave background and the nearer universe observed by today's spaced based observatories as illustrated in figure 1.

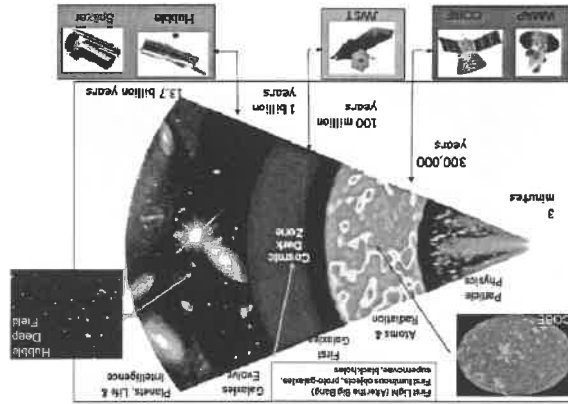


FIGURE 1. History of the Universe and Observation Capabilities of Telescopes

JWST (Figure 2) has a 6.5 meter primary mirror, with a sun shade the size of a tennis court that will protect the observatory from the heat of the sun, allowing it to operate as cold as 32K to

92% by machining an isogrid on the back surface. The operating temperature is 32-59K. The maximum surface figure error of the mirror is 25 nm rms.

CRYOGENIC ACTUATOR

Ball Aerospace developed a cryogenic linear actuator to drive the hexapod and radius of curvature systems on the primary mirror segments [1]. This actuator is capable of operating at temperatures down to 20K and has a step size of < 10 nm and a range of 20mm. The very small step size over such a large range is achieved by using a two stage actuator, driven by a single motor. The fine stage of the actuator is based on a flexure reducing motion structure patented by Ball Aerospace, patent number 5,969,892. This device was designed to convert relatively coarse motion to optically precise fine motion with minimal backlash or hysteresis. This patented design was refined for use on JWST as shown in figure 4.

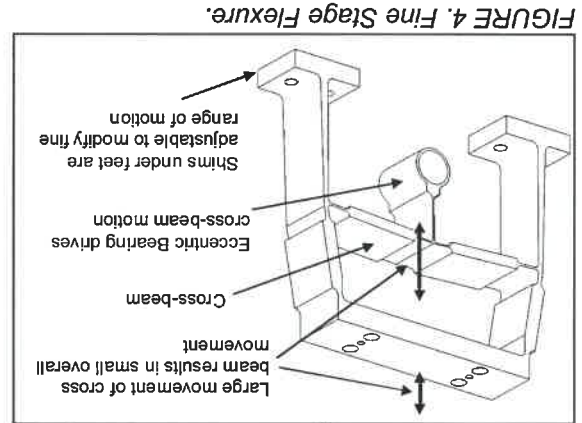


FIGURE 4. Fine Stage Flexure.

This fine stage flexure provides a 10 micron range of motion with 10 nm step size. For motion beyond 10 microns the coarse drive is employed by turning the motor to engage an idler gear and drive a ball screw. The ball screw has a total range of 21 mm with approximately 60 nm step size.

Figure 5 shows a schematic of the actuator gear train. A stepper motor drives a 60:1 gear head that drives a 3:1 spur gear pass. The shaft attached to the spur gear drives the cam that actuates the fine stage flexure. It also drives a 1:1 bevel gear that moves the idler gear. When the idler gear is disengaged only the fine stage is driven. When the idler gear engages it actuates a 8:1 spur gear pass that drives a 2mm pitch ball screw to produce coarse range. After reaching the desired coarse stage length, the

motor is reversed, disengaging the idler gear, and allowing for fine stage adjustment.

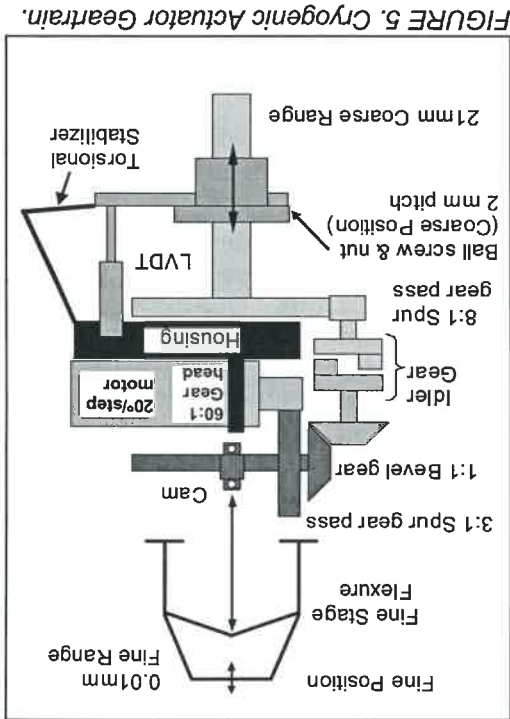


FIGURE 5. Cryogenic Actuator Geartrain.

ASSEMBLY REPEATABILITY

The precision polishing for the JWST primary mirrors to achieve a 25 nm rms surface is a wet process that the actuators cannot be exposed to. Using reasonable manufacturing tolerances does not allow the actuator system to be installed without deforming the mirror.

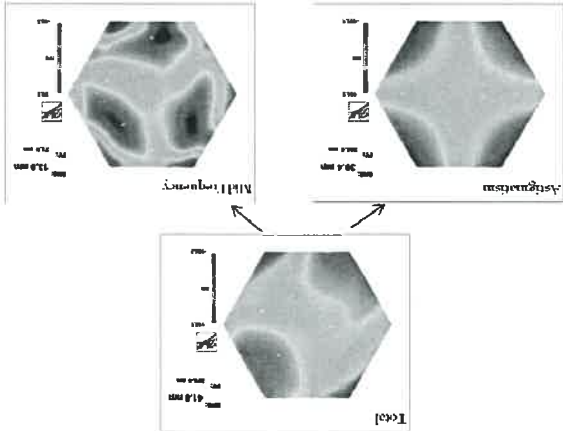


FIGURE 6. Surface figure deformation due to actuator assembly.

This constraint led to the development of a manufacturing process that would protect the actuators from the polishing requirement but imposed a stringent design requirement to ensure that assembling the actuators would

produce the same deformation in the mirror each time.

Manufacturing Process

A manufacturing process was developed to fabricate the mirrors efficiently at multiple facilities, many built for the sole purpose of manufacturing JWST mirror segments.

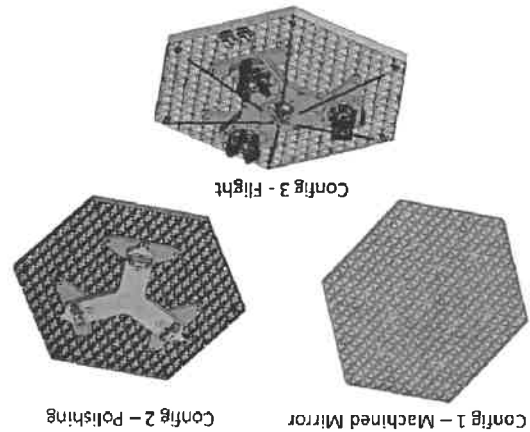


FIGURE 7. Mirror configurations for manufacturing processing.

The mirrors were processed in three different configurations. Configuration 1 is the bare mirror with no structure attached to the back. The initial machining operations to fabricate the mirror are obviously performed in this configuration, then initial polishing to 150 nm rms is also performed in configuration 1. After this, the mirror is shipped to Ball Aerospace for assembly to configuration 2. Configuration 2 simulates the actuator support on the mirror which provides a more robust mount on which to optically measure the mirror. The mirror is then assembled to configuration 3, the final flight configuration with all actuators mounted to the mirror. The difference between these two configurations creates the assembly target map shown in Figure 6. The mirrors are then characterized for the subtle warping that occurs between room temperature and cryogenic operating temperature to develop a cryogenic target map. After these measurements occur the mirror is reconfigured to configuration 2 for final polishing which accounts for both the assembly and cryogenic target maps. The mirror is then gold coated and re-assembled to configuration 3, the final flight configuration. Acceptance testing is then performed to verify that the mirror meets all requirements. It is crucial that when the mirror is re-assembled to configuration 3, that the assembly deformation

remains the same as when first characterized since the inverse of this deformation has been polished into the mirror.

Designing for Repeatable Assembly Deformation

The primary mirror segment is a whiffle tree design with 9 mounting locations on the mirror that are attached together as 3 sets of 3 mounts via whiffles. These 3 whiffles are then attached, via the hexapod, to a single delta frame. This delta frame is then attached to the telescope structure using 3 points. Each mount uses a quasi-kinematic approach employing a flexure with high stiffness in radial and axial directions and low stiffness in the remaining 4 degrees of freedom. The 2 degree of freedom restraint in three locations restrains the requisite 6 degrees of freedom without over constraint. The radius of curvature system can be considered a separate structure that rests on the back of the mirror. This system also uses a quasi-kinematic mounting approach, although more complex to describe.

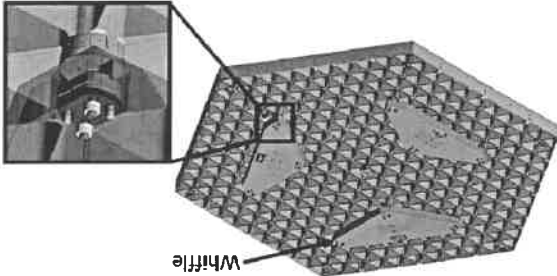


FIGURE 7. Mirror with whiffles attached. Detail showing 1 of 9 flexures that attach whiffle to mirror.

Each structure attached to the mirror was evaluated to ensure that the required assembly deformation would be met. The first level of isolation, the whiffle to mirror, was very sensitive, requiring less than 0.25 microns stability at the joint. It was determined that this joint should be bonded to the mirror and then bolted and pinned to the whiffle, with the pins potted into place to achieve the required stability. This attachment was then permanent and this structure remained in place on the back of the mirror through all polishing activity in configuration 2. Reduction in sensitivity at this joint by reducing flexure stiffness was not feasible without compromising the ability of the system to withstand launch loads.

The hexapod system, containing the 6 actuators that allow for 6 degree of freedom movement of the mirror, had to be removed for polishing. This system needed to have very low stiffness flexural joints to allow the hexapod to move the mirror and this requirement naturally provided a low stiffness attachment to the mirror. This interface required only 30 microns of stability which was easily provided by a hole and slot pair which slip fit over two pins. This approach created a precision interface without the need for costly match machining.

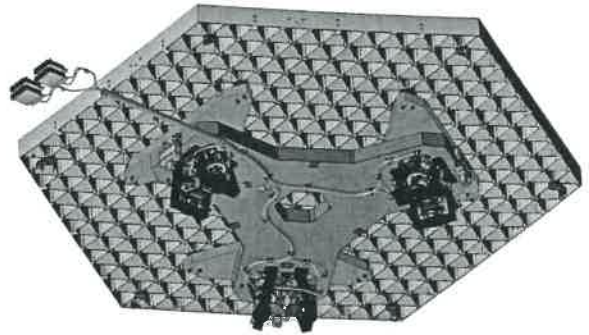


FIGURE 8. Mirror with hexapod attached.

The radius of curvature system, shown in figure 9, also needed to be removed for polishing. This structure was evaluated to determine which interfaces should be dis-assembled and which needed to remain together.

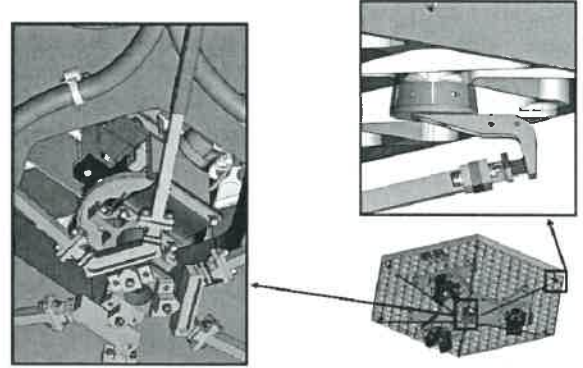


FIGURE 9. Radius of curvature actuation system.

The 6 struts running to the edges of the mirror were designed for low sensitivity and required only 30 microns of stability. This joint was made using a custom designed precision shoulder screw that could act as both a shear pin and provide the requisite clamping force for the joint. The actuator attachment to the center of the mirror was the most significant design challenge.

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ACKNOWLEDGEMENTS

Work was supported in part by the Ball Aerospace & Technologies Corp. subcontract under the JWST contract NAS5-02200 with NASA GSFC. The JWST system is a collaborative effort involving NASA, ESA, CSA, and the astronomy community. We acknowledge the support and contributions of the many engineers and scientist among the many organizations that make up the JWST program.

CONCLUSIONS

This part was highly stressed due to the launch environment and the rotational stiffness of the interface had to be increased substantially to obtain adequate strength. The result was an interface with a stability requirement of 5 microns. This interface also used a hole and slot pair to attempt to provide the required stability. This interface proved to be challenging during assembly and required re-work at times to achieve the required mirror surface deformation confirmed through optical measurements. The assembly technicians created detailed procedures for assembling this joint that biased the joint at assembly to try to minimize variability and then used a specific torquing procedure for the bolts to obtain the best joint repeatability. Implementation of this procedure greatly improved the repeatability of the joint and produced repeatable assembly deformation.

ALIGNMENT FIXTURE FOR KINEMATIC MOUNTS OF SCIENTIFIC INSTRUMENTS ON HUBBLE SPACE TELESCOPE

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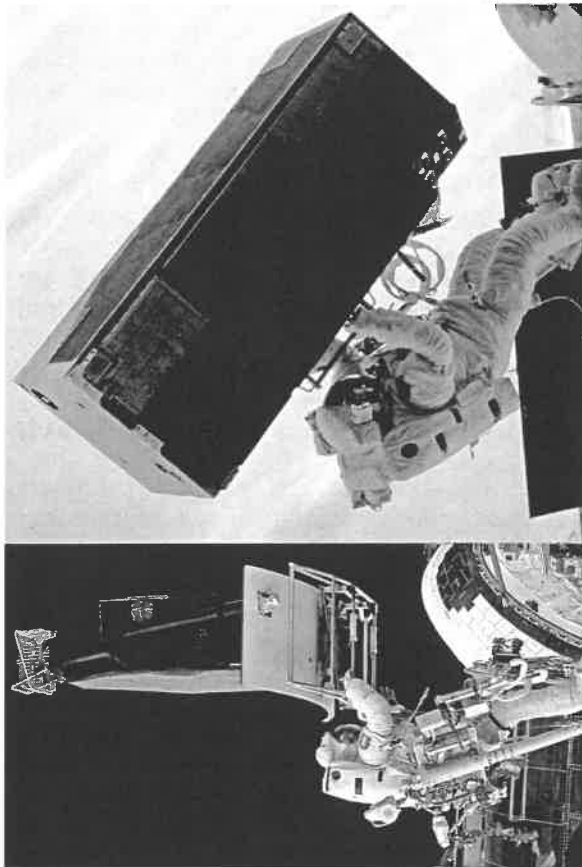


FIGURE 1. The first servicing mission to HST in 1993 proved the importance of the replaceable design enabled by PAAF. The new instruments used optics to correct the spherical aberration in the primary mirror. [Photo credit: NASA]

During initial positioning in the Hubble focal plane, the position of each instrument must be known to better than $10\text{ }\mu\text{m}$ [4]. When replaced in space, the instrument position must repeat to better than $15\text{ }\mu\text{m}$. These requirements were met using kinematic mounts which serve the additional function of insulating the instrument from heat transfer through the structure. Each of the eight Points A latches constrains three translational degrees of freedom. Two more latches, Points B and Points C, constrain the remaining degrees of freedom.

ABSTRACT

The Hubble Space Telescope (HST) has revolutionized astronomy since its launch in 1990 and through new instruments in four servicing missions since. A critical design feature that enabled the "fix" in 1993 and several upgrades is the ability to replace science instruments in space. The precise positioning requirements are realized using kinematic latches. This paper describes the design features of an air bearing cylindrical coordinate measuring tool used to locate the eight critical "Points A" locations for HST. This Points A Alignment Fixture (PAAF) was essential during telescope assembly in 1985 to position the kinematic mounts inside Hubble. PAAF was again called upon in 1993 to re-create the as-built flight hardware for the ground-based High Fidelity Mechanical Simulator [1]. The PAAF incorporates several precision engineering aspects to locate the scientific instrument mounts to the micrometer-level.

BACKGROUND

Considered by some to be the most successful space telescope ever built, HST is the first observatory designed for maintenance and repair while in orbit [2]. Since Hubble orbits above Earth's atmosphere, spectral absorption and turbulence distortions are avoided. HST was launched in 1990 and consists of a Ritchey-Chrétien telescope, support trusses, and focal plane structure with five scientific instruments and three guidance sensors. The longevity of the HST is due to several successful servicing missions to enhance and repair equipment such as the science instruments shown in Figure 1. The design allows astronauts in bulky, pressurized spacesuits to replace all of the eight instrument packages [3].

There are two removable arms for setting the Points A balls—the Radial A Fitting Gage (RAFG) and the Axial A Fitting Gage (AAF). The arms are removable to allow the gage to rotate from one ball to the next, maneuvering around the struts of the Focal Plane Structure of HST.

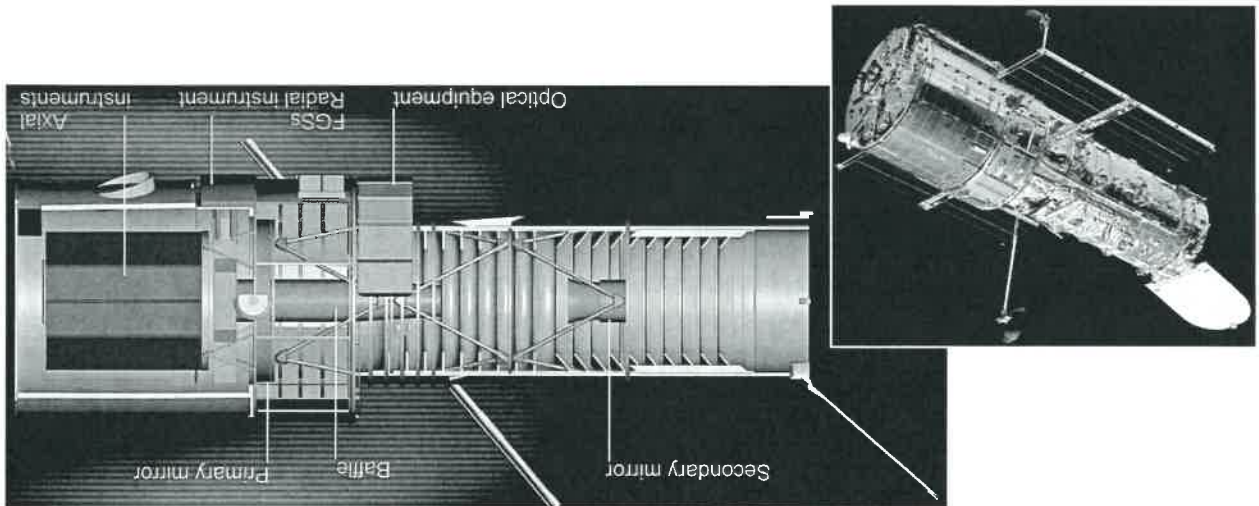
The spindle rotor is held at the desired orientation angle using a pneumatic locking feature. The internal porting is split so that one thrust and radial bearing is supplied independently of the other thrust and radial bearing. Air pressure on one side is lowered while maintaining full pressure on the other side. This feature is used as a non-influencing metrology brake since the thrust bearing with the higher pressure shifts axially 5 μm due to piston force and the opposite thrust bearing is pushed against the stator.

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PAAF DESIGN

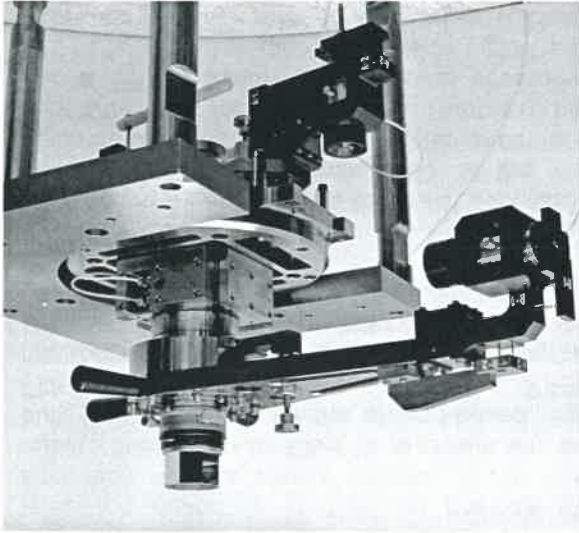
PAAF, shown in Figure 3, located the critical Points A balls—four mount at a common axial location and four mount radially. At the heart of the PAAF is a Professional Instruments Model 4B BLOCK-HEAD[®] air bearing spindle. The spindle was aligned with the optical axis during assembly of the telescope. This spindle has less than 25 nanometers radial and axial error motion and less than 100 nanoradians tilt error motion.

FIGURE 2. Sketch showing the optical telescope assembly with instrument packages. Light enters the aperture and is reflected by the 2.4 m diameter primary mirror (concave asphere) to the 0.3 m diameter secondary (convex asphere). The light then travels through a hole in the center of the primary mirror to four axial scientific instruments, one radial scientific instrument and three fine guidance sensors. [Photo credit: NASA/ESA]



Both arms use a light interference-fit (approximately 1 micrometer interference) spherical pilot mount at the interface with the spindle which ensures repeatability of better than 0.5 μm .

FIGURE 3. Points A Alignment Fixture used to precisely locate eight balls in three dimensions for the HST instrument packages. PAAF is a cylindrical coordinate measuring tool with removable arms, two linear air bearings, and a rotary air bearing spindle.



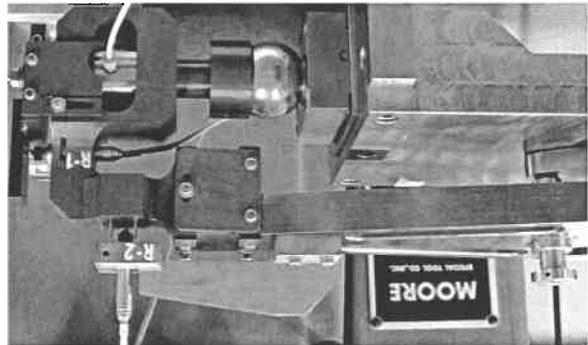


FIGURE 4. Close-up view of the RAFG of PAAF. A linear air bearing with a conical cup seats on the ball. The null position was found by slight rotation of the arm.

Figure 4 shows a close-up of the conical cup at the end of the radial arm that seats on the radial Points A balls. An eccentric cam provides a slight rotation of the arm for nulling. A linear flexure provides axial nulling. LVDTs are used for radial and axial displacement measurements. Polygons are attached to the air bearing spindle collinear to the telescope optical axis to be used with an autocollimator for angular position feedback.

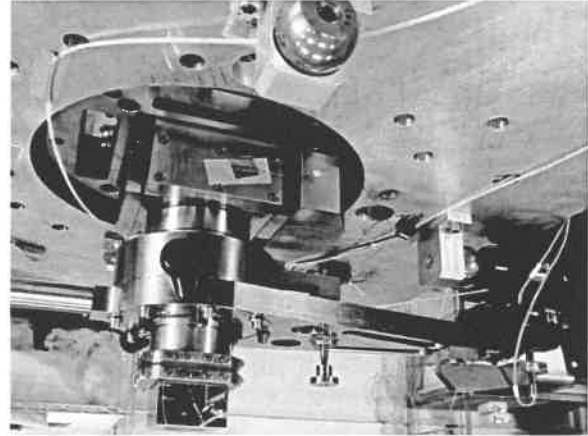


FIGURE 5. PAAF being used to position the radial balls. [Photo credit: Perkin-Elmer]

PAAF MASTERING

Calibration of the mastering tool for PAAF was accomplished on a modified Moore No. 3 Jig Grinder. The Moore machine was used as a very accurate three-axis motion stage. The linear positioning accuracy of the travel was certified using a Moore 18 inch step gage. Rather than use the existing grinding spindle, a

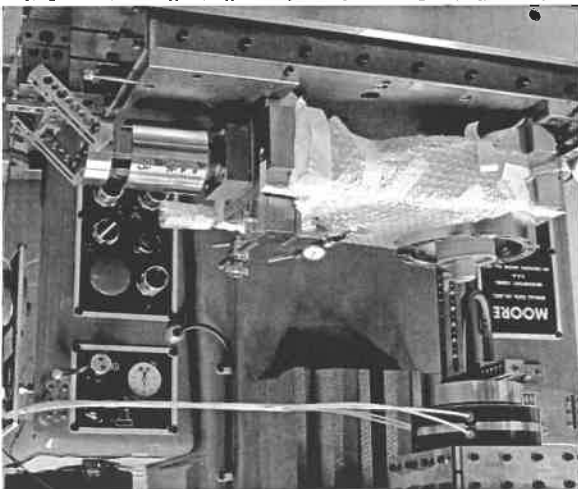


FIGURE 6. Sweeping the pilot diameter of the RAFG mastering tool using a Mikrokator.

Professional Instruments Model 4R BLOCK-HEAD® air bearing spindle was mounted to the quill of the jig grinder. The axis of rotation of the spindle was adjusted to be perpendicular to the XY-plane of the Moore No. 3. The spindle also used a pneumatic lock while indicating various features of the mastering tool. The approach master tool is shown in Figure 6. A CE Johansson Mikrokator with 250 nanometer resolution was used to align and locate the pilot diameter of the mastering tool. The distance from the pilot to the mastering ball was certified using the Moore X-axis slide.

With the mastering gage approved, the LVDTs of the RAFG were zeroed as shown in Figure 7. RAFG is inserted into the pilot of the mastering tool and the linear air bearing with conical receiver cup was seated on the ball. The null position was found by scanning the cone over the ball and minimizing the LVDt output.

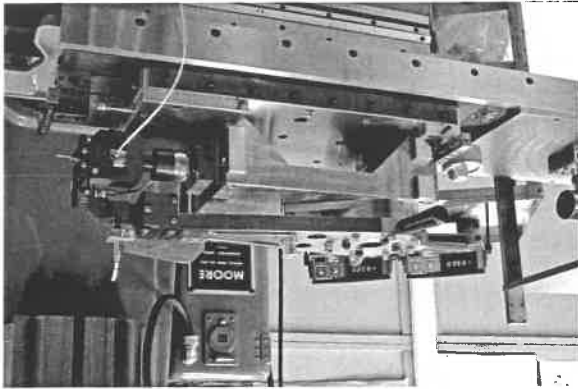


FIGURE 7. Zeroing the LVDTs of the RAFG.

Next, a polygon was installed on the RAFG while it was mounted in the mastering tool as shown in Figure 8. The polygon was aligned parallel with the gaging datum of the mastering tool. For final assembly, the polygons were attached with a hard mount.

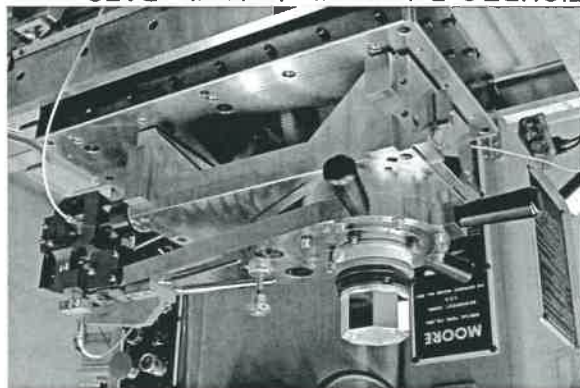


FIGURE 8. Polygon attached to the RAFG.

Figure 9 shows part of the process of aligning the polygon to the datum. First, an autocollimator was oriented normal to the reflective datum. Then a large plano mirror was placed between the datum and the first autocollimator. A second autocollimator at the height of the polygon was adjusted normal to the large mirror. Finally, the mirror was removed and the polygon orientation was adjusted normal to the second autocollimator.



FIGURE 9. Procedure for aligning the polygon on the RAFG relative to the reflective datum on the mastering tool.

CONCLUSION

PAAF was a crucial element to achieve the precision required for the HST Points A latch positions. The PAAF was also instrumental in establishing the Points A latch locations for the ground-based High Fidelity Mechanical Simulator which has been used for every instrument since the first servicing mission in 1993. The high accuracy and repeatability of the air bearings in PAAF combined with remarkable precision engineering have contributed to Hubble's continued success.

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Nickel Plated Metal Mirrors for Advanced Applications

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INTRODUCTION

Mirrors made of a metallic bulk material plated with electroless nickel as a polishing layer are widely used in optical systems [1]. An athermal design can be achieved if both, the optical element and the structure are based on the same substrate material such as Al6061, aluminum-silicon alloys or metal composites. Furthermore metal mirrors and structures are cost efficient because the substrate material offers excellent machinability. A direct integration of mounting and reference structures for a system assembly without time consuming alignment steps is also possible [2]. A better understanding of critical material and process issues such as dimensional stability or the removal characteristics during polishing and figuring can enlarge the scope of nickel plated metal mirrors towards high performance applications. In particular reflective optics for short wavelengths and optical systems working under cryogenic conditions are of interest [3].

DIMENSIONAL STABILITY

Matching the CTE

In addition to the space qualified Al 6061 as a standard, a variety of materials can be utilized as a bulk material for Nickel plates mirrors. Besides the higher specific stiffness the main considerations while choosing a proper substrate material shall address the harmonization of the thermal expansion.

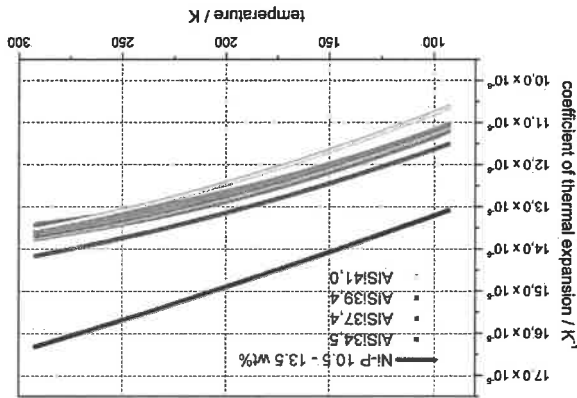
Material	Density [g/cm ³]	Young's Modulus [GPa]	CTE @ 20 °C [K ⁻¹]	Therm. conductivity [Wm ⁻¹ K ⁻¹]
Al6061	2,7	70	23 x 10 ⁻⁶	160
AlSi(40)	2,54	102	13,4 x 10 ⁻⁶	150
AlSiC(30)	2,91	117	12,4 x 10 ⁻⁶	123
AlBeMet	2,07	193	13,9 x 10 ⁻⁶	210
Be	1,85	287	11,3 x 10 ⁻⁶	190

FIGURE 1. Metal bulk materials for electroless Nickel plated mirrors.

In order to maintain considerable advantages like material and machining costs, safety in terms of

material-compositions.

FIGURE 2. Measured CTE curves for different

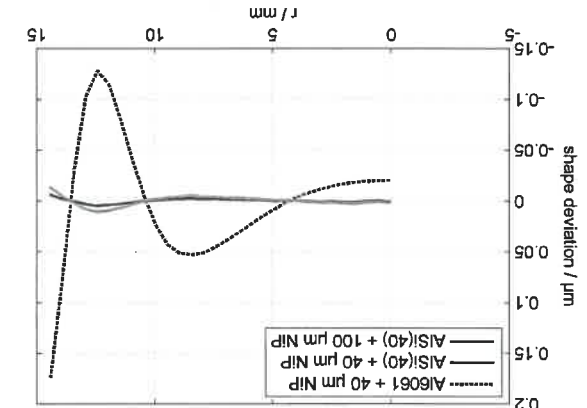


Similar to the ratio by mass of aluminum and silicon (substrate material), the phosphorous content influences the CTE of the polishing layer. State of the art galvanic processes permit the deposition of electroless nickel with a phosphorous content ranging between 10.5 – 13 wt %. The minor influence of variations in the phosphorous content on the CTE is also shown in figure 2. Nevertheless, an amorphous nickel with a controlled phosphorous content can be coated by controlling the pH-value

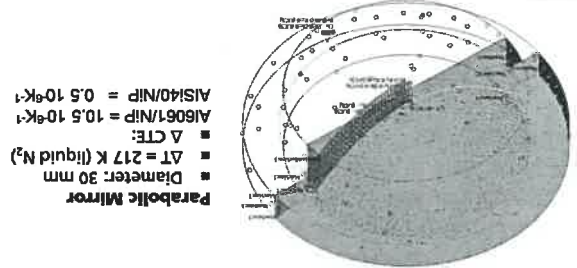
As shown above finite element methods or analytic calculations are suitable to predict the structural mechanic behavior under the influence of temperature and static or dynamic forces. The modeling is based on the knowledge about the mechanical and thermal properties such as the specific stiffness or the CTE. A better knowledge of the change of these values as a function of the temperature is mandatory for a proper and accurate modeling. Intrinsic substrate stress and residual stress, which are caused by machining or coating processes, are hardly to quantify. Residual stress is critical regarding thermal and time driven relaxation of the material that causes a change of the figure. However these stresses are often neglected in FEM or analytical simulations. Thus the impact of the thermal cycling and annealing on

Stress relief procedures

FIGURE 3. FEM-model and temperature motivated shape deviation of a parabolic mirror.



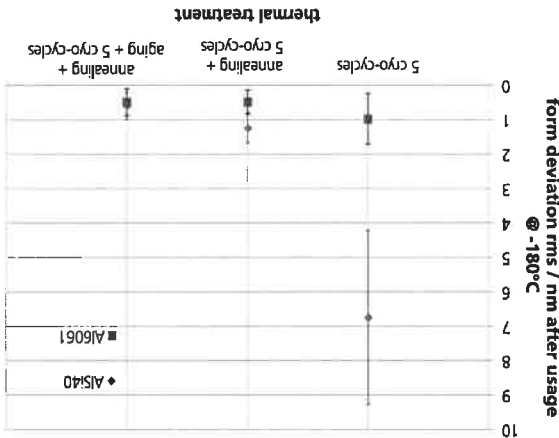
Shape Deviation from Any Best Fit Parabolic Al-Alloy Based Mirrors due to Cooling to Cryogenic Temperatures



and nickel concentration of the electrolyte during the deposition. Hence a second key for the CTE harmonization is provided. By matching the CTE of the substrate and the polishing layer, the bimetallic bending can be reduced significantly. The impact of the CTE harmonization to avoid bimetallic bending is shown below with an example of a parabolic mirror for operating conditions at 77 K. In comparison to Al6061 as a substrate the FEM simulations show a significant lower deviation from the best fit parabola at operating conditions.

Metal mirrors offer a fundamental advantage: diamond cutting allows a flexible and efficient shaping of sophisticated optical surfaces such as off-axis aspheres or freeforms. Focusing on applications in the visual and ultraviolet wavelength range, the surface figure as well as microroughness and straightness are of crucial importance. Typical specifications for a clear aperture (CA) of 100 mm are figure irregularity less than 15 nm (root mean square - rms). The microroughness shall not exceed values between 0.5 nm to 0.2 nm (rms) for high spatial frequency roughness (HFSR). To meet these requirements, subsequent post polishing and local figuring techniques are mandatory. Furthermore the diamond cutting has to be improved to avoid mid special frequency errors. Diamond turning usually provides figure accuracies between 0.5 - 1 µm peak to valley (p-v) for an aspherical mirror with a CA of 200 mm. The residual form error typically consists of a non-rotationally symmetrical error

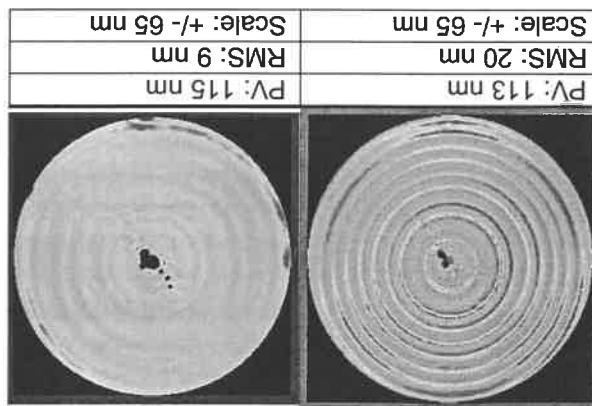
FIGURE 4: Improvement of dimensional stability due to thermal treatment.



the relaxation of residual stress has to be examined in experiments. Currently a study of stress relief procedures for nickel plated aluminum alloy substrates with a silicon content of 40% is conducted. First results indicate, that primarily internal stress on the microscale between the material phases of the substrate has to be relieved. In comparison to Al6061 the number of thermal cycles for stress relieving and aging has to be increased. For the cryogenic usage a thermal treatment down to the operating temperature is mandatory for at least 5 times. Figure 4 illustrates the stress reduction by different thermal treatments of flat sample mirrors with a CA of 48 mm. The graphs show the remaining form deviation after the usage at operation temperature (-180 °C).

Electroless Nickel coatings can be figured with sub-aperture techniques like Ion Beam figuring (IBF), Magneto Rheological Finishing (MRF), computer controlled pad polishing (CCP) or Fluid-jet polishing (FJP) /4/, because of the hardness (> 500 HV), the amorphous structure and the non-magnetic behavior of the material. IBF is used for the figure correction only and enables the best results in terms of figure accuracy. MRF, FJP and CCP allow also a smoothing of the microroughness, while the shape is improved. The typical diamond turning marks with a high spacial frequency of some microns can be effectively removed and randomized. CCP as a traditional polishing method leads to a superior surface roughness which enables the use of Nickel plated mirrors in the UV and EUV spectral range. Typical values for a CCP

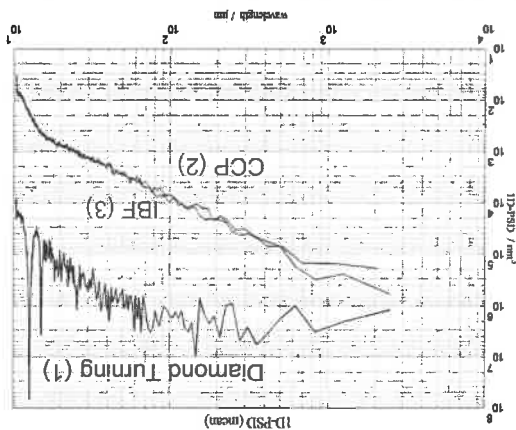
FIGURE 5: Decreasing of mid-frequency errors of a freeform mirror, local figuring technique: Ion Beam Figuring, CA: 205 nm.



map and a rotationally symmetrically share which is directly related to the diamond turning process. The non-rotational residuals provide the dominant contribution to the PV value after diamond turning. These are typically long-wave errors and have a good response to local finishing techniques. The rotationally symmetric shape errors occur with a periodicity between 0.1 mm to some millimeters. Although they show small amplitudes of some tens of nanometers, they are difficult to correct. Therefore periodic fluctuations caused by temperature changes, fluctuations in spindle air pressure or drive elements have to be avoided as much as possible. Figure 5 illustrates, by the example of a freeform mirror with 205 nm in diameter, the reduction of mid frequency errors as result of an improved temperature control (+/- 1K vs. +/- 0.2 K). Analyzing the rms and p-v values it becomes obvious that rms tolerancing makes much more sense to ensure the optical function.

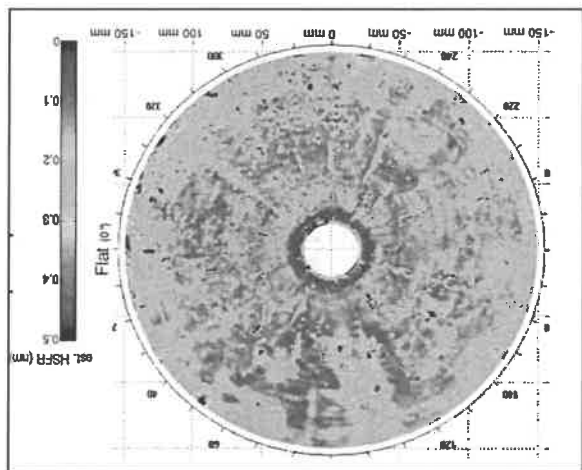
MRF offers the advantage to smooth the surface texture but to correct the surface figure at the same time. MRF technologies are of special interest regarding applications in the near infrared. For NIR wavelengths, shape accuracies < 20 nm (rms)

FIGURE 7. PSD for a Nickel plated surface after different manufacturing processes (measurement field 2.5 mm x 2.5 mm, white light interferometer).



Using the example of an asphere with 300 mm diameter it is demonstrated, that Nickel plated metal mirrors can be super polished to less than 2 Angstrom (rms) in the HFSR range (Figure 6). For imaging systems where both, high shape accuracy and roughness will be a key figure, a combination of CCP and IBF is preferred to improve the diamond turning results significantly.

FIGURE 6. HSFR mapping of a, aspherical Nickel plated mirror (300 mm diameter) determined from light scattering measurement. Average HSFR 0.14 nm (rms).



roughness are: < 0.5 nm rms for a measurement field of 2.5 mm x 2.5 mm, < 0.3 nm rms for a measurement field of 150 µm x 150 µm.

have to be achieved and the periodical diamond turning tool marks have to be removed to avoid diffraction. The microroughness specification is less challenging (< 5 nm rms) for these spectral ranges and can easily be achieved using MRF technology. Under these conditions, MRF can make use of its inherent advantage: the simultaneous and efficient improvement of surface form and roughness. Figure 8 demonstrates the improvement of shape accuracy using MRF for a flat mirror. The required process time for the correction cycle was 10 minutes. With the same polishing step the micro roughness was decreased from 7.3 to 1.6 nm (rms) @ 40x magnification (180 x 130 µm measurement field).

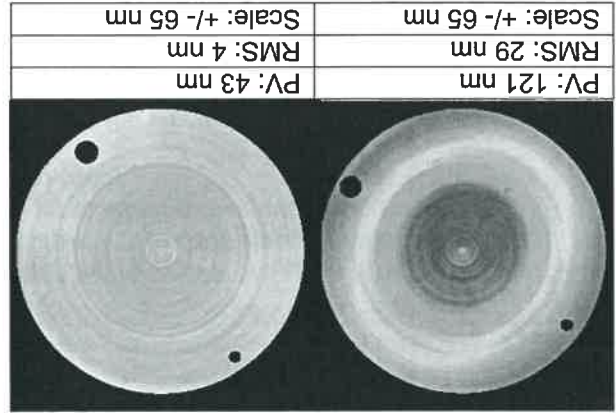


FIGURE 8: Deterministic figure correction of a flat Nickel plated mirror, local figuring technique: MRF, CA: 48 mm.

CONCLUSION

By applying deterministic polishing techniques nickel plated metal mirrors can reach shape and roughness performances comparable to ceramics (SiC), glass ceramics or glass (ULE™, ZERODUR®, borosilicate glass). By the usage of a CTE-tailored material such as AlSi40 the bimetallic bending as a main impact on dimensional stability over temperature can be significantly reduced. Concerning mechanical properties and the causal low thermal distortion mirrors made of SiC or glass ceramics are preferred for challenging applications [5]. However metal mirrors provide a couple of advantages regarding the system alignment and integration. Well established CNC-machining and diamond cutting allows a great deal of freedom for:

- Direct integration of reference structures and alignment marks
- Direct integration of elements for kinematic decoupling
- Direct integration of cooling or heating channels

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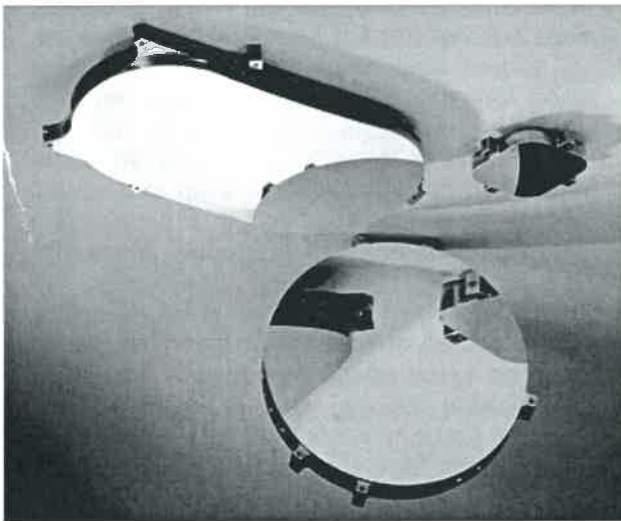
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ACKNOWLEDGEMENT

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FIGURE 9: Nickel plated mirrors for a multispectral imaging radiometer (substrat material AlSi40).



- Flexible lightweighting.

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Abstract

The design of optical concentrators for solar systems poses unique engineering challenges.

In comparison with precision optical instruments, solar concentrators are crude devices. However, they are also high-volume systems that are subject to strict budget constraints. Solar concentrators are also required to last 20 years in outdoor service with minimal maintenance.

The intersection of these requirements is very demanding, and has been the subject of much design work over the last several decades. Current designs fall short of simultaneously satisfying all of the precision, longevity and budget requirements.

This paper concentrates on these requirements as they pertain to parabolic dish concentrators, and illustrates how following precision engineering principles helped us achieve both cost savings and performance enhancements.

Selection of Concentrator Type

Solar concentrator geometries include (among others) parabolic troughs, linear Fresnel systems, heliostat fields, parabolic dishes.

The dish geometry offers the best performance of all concentrator geometries, but is traditionally the most expensive and difficult to construct.

The advantages of dishes include the highest concentration factor, the highest reflector utilization, the best terrain flexibility, and ability to work with almost any generation technology, including PV, Stirling, Brayton, or Rankine.

The disadvantages include high cost and installation time — enough to offset the advantages. No solar dish company has been successful to date.

Our rationale in choosing the dish geometry was that it appears more feasible to make dishes affordable and practical than it is to make the other system types perform better.

The type of powerhead also affects the requirements levied on the primary optics.

Thermal receivers are generally less sensitive to illumination variance, and their cost is largely a function of their power rating, not their aperture area, and so the use of more aggressive under-filling is allowed. They are also able to time-average fast-changing optical effects, relaxing the requirements on the tracker.

Photovoltaic receivers, on the other hand, are highly sensitive to illumination variance, and their cost is directly proportional to their area. These issues are partially alleviated by the common use of a homogenizing secondary optical element situated behind the aperture. PV heads typically operate at a lower concentration factor than their thermal equivalents.

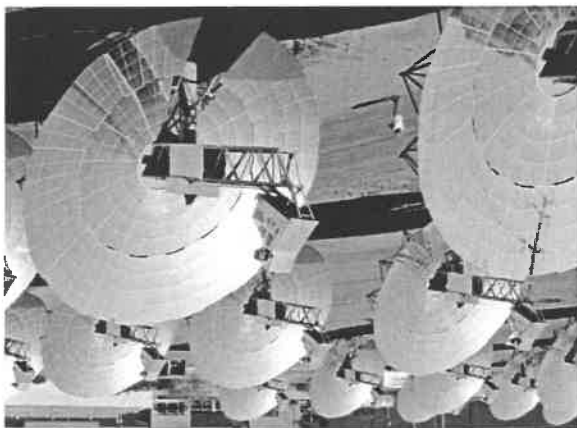


FIGURE 1. Pilot field of sixty 12 m dishes by SES (Stirling Energy Systems)

Size and Budget

To be economically viable, a 12 m diameter dish such as shown in Figure 1 has to be manufactured for 25,000 — 35,000 US dollars, including power conversion unit, tracker, ground foundation, and installation. To directly compete with flat-panel PV, it needs to cost about 50,000 — 75,000.

Correspondingly, manufacturing volumes are high. A 1 GWatt-peak field requires 40,000 such dishes. In order to deploy 1 GWatt/yr, the dishes should be manufactured and installed at a rate of four dishes per hour, around the clock.

For comparison, world demand is increasing by 100 Gwatt per year, Total installed PV capacity is now in the 10s of GWatts.

Error sources

The requirement for all concentrator systems is to divert the collected light into the aperture.

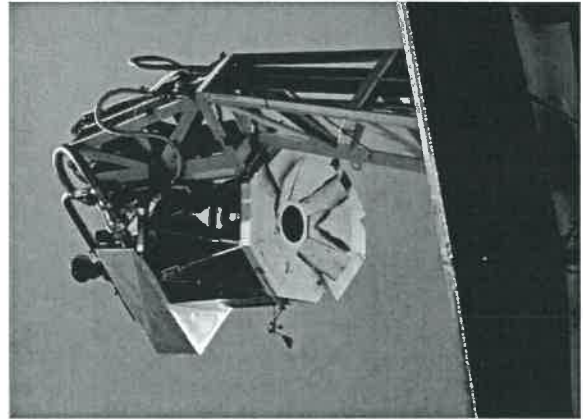
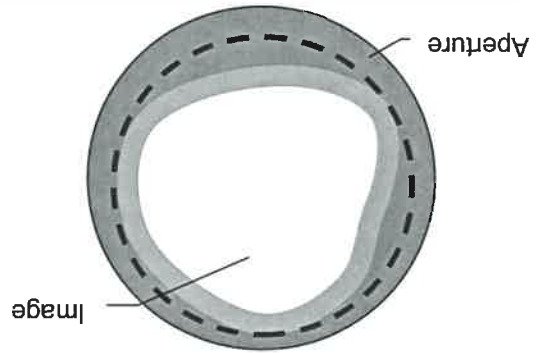


FIGURE 2. Schematic diagram of the focal image inside an aperture, and a photograph of a real system (Courtesy Herbert Hayden, Southwest Solar Technologies).

The situation shown in Figure 2 (top) illustrates the designer's dilemma. The focal image is deformed, non-uniform, and has a low-intensity

edge (a result of reflector shape inaccuracies, defocusing, and the sun's intensity profile). It is also offset from the center due to tracking errors. If we consider a parabolic dish with $f/D=0.6$, and a desired 2000x concentration over a flat aperture, the apparent radius of the aperture from the edge of the dish is roughly 11 mRad. (and somewhat larger from its center). For a parabolic dish, a typical error budget is comprised of the following items, shown along with near-term goals for the state of the art: (figures are half-angles)

- Reflective surface non-specularity (85–90% reflectance within a 2 mRad cone)
- Reflector tile geometry aberration (1 mRad)
- Reflector tile positioning error (1 mRad)
- Powerhead positioning error (1 mRad)
- Tracking error (2.5 mRad)
- Sun radius (4.5 mRad)

Thus in principal it is just possible to meet the required focal image size.

Beyond this general discussion, for each particular system, it is not enough to compute an RMS of the errors, but detailed simulations and measurement of the interactions of these factors are undertaken.

Andraka et. al. performed an in-depth study of effects of facet misalignment on illumination patterns inside the Stirling Energy Systems (SES) thermal receiver. They have also developed an impressive optical alignment measurement tool, detailed in [2], and a simulation tool modeling these effects.

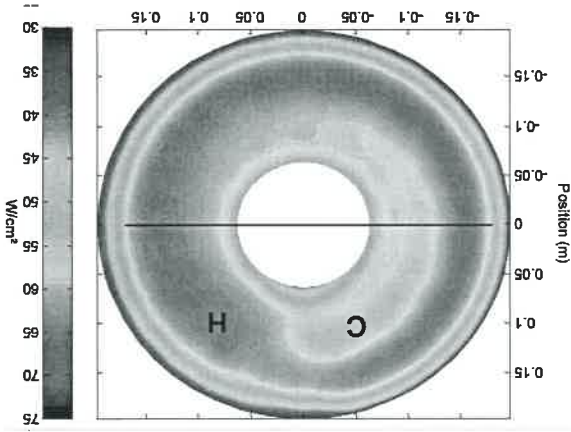


FIGURE 3. From [2] – The effect of a 1.5 mRad facet "twist" misalignment (with no tracking error) on focal image, showing hot (H) and cold (C) spots. (Courtesy Charles Andraka, Sandia National Laboratory)

The common philosophy in both dishes is to base the dish on a truss which is built to be load bearing but not overly accurate. With the structure in place, reflector tiles are attached to it using 3-point adjustable mounts. And then tweaked into place, guided by an optical reference.

A second approach is the "accurate as assembled" construction of smaller dishes such as Infinia Systems'. These dishes are more expensive per unit area, and the designs do not scale up to larger sizes.

Alignable Carrier Structure

Heliocentric's approach is to make the carrier truss precise enough so that tile alignment becomes unnecessary.

There are several classical reasons trusses are not accurate "as built". First, even though some trusses are kinematic in principle, they are never constructed this way, and so the structure is not definite and contains internal stresses. Second, the large part count introduces long tolerance stack ups, and finally, the assembly joints are not precise, so assembly tolerances add up on top of part tolerances, exacerbating the previous two issues.

To eliminate these causes, we introduced a kinematic truss, familiar to most people as a tensile-spoke bicycle wheel, as the basis of the dish. The advantages are immediate: The structure is kinematic and rigid, and weighs only 25% of a comparable standard truss. Most components are loaded in a fixed direction (either tension or compression), and all assembly joints are uniquely loaded and thus have zero play.

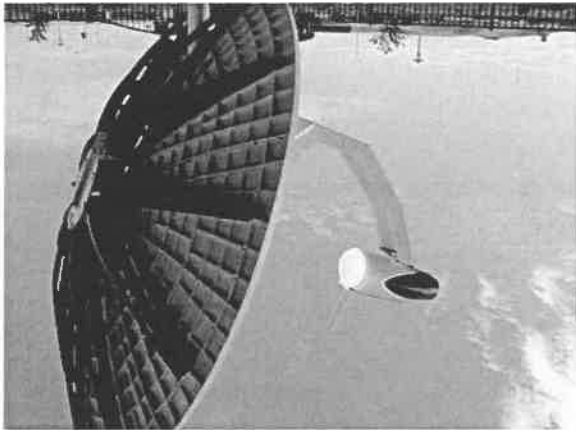
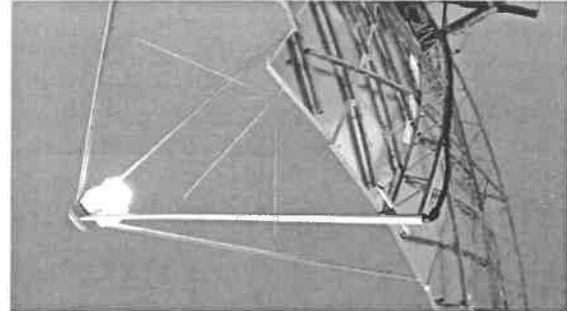


FIGURE 6. Infinia Corp's 17 m² linear Stirling dish

FIGURE 5. Concentrating dishes: With a thermal powerhead by Stirling Energy Systems (top) and photovoltaic by Solar Systems (bottom)



Two 100 m² state of the art dishes are shown below – one by Stirling Energy Systems, and one by Solar Systems.

Standard Constructions

Alternatively, laser based instruments such as NREL's VSHOT are used to map the large-scale geometry of the dish, eliminating the effects of reflector surface imperfections.

FIGURE 4. Mapping the flux near the focal plane. The high light intensity makes measurement difficult. A common technique is to image the full moon instead. (Courtesy Southwest Solar)



In this figure, the focal image is about 90% of the aperture, leaving 5% in each direction for tracking errors.

Structural alignment is achieved by tweaking the spoke length, which can be done to almost arbitrary precision. Since assembly tolerances were taken out, it is possible to use "yielded to length" spokes, which can be fabricated to μm -level tolerances with very little effort.

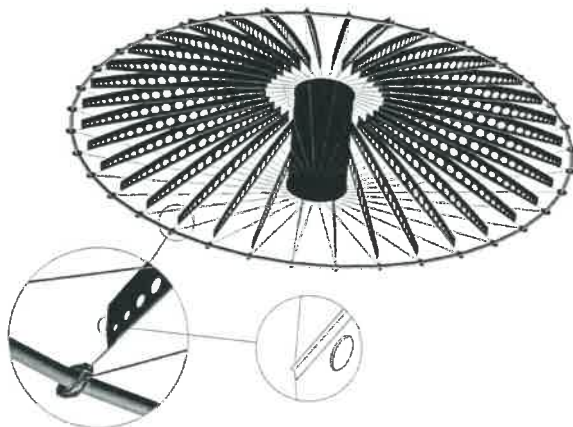


FIGURE 7. Tensile kinematic alignable truss with spoke stiffening ribs

If using 36 spokes, the structure has 108 DOFs and 108 constraints, 72 of which are adjustable, (2 per tensioning node – axial and radial)

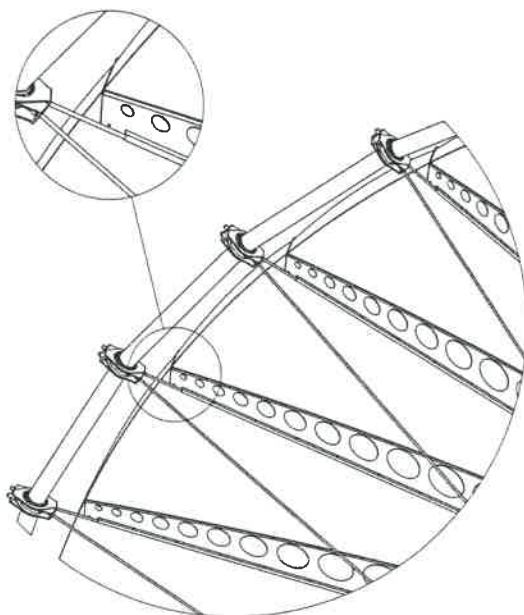


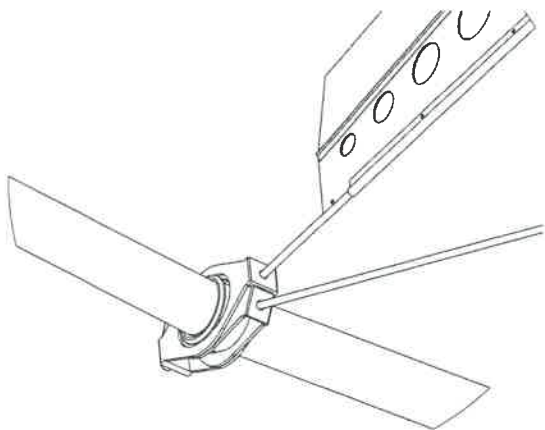
FIGURE 8. Truss and tiles. Reflectors tiles are attached from the back side and their optical surface is positioned by the ribs.

The reflector tiles attach to curved rulers that are pre-attached to the stiffening ribs, from the back side of the dish. The advantage here is that it is the front of each tile that is being positioned,

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FIGURE 9. 2-DOF Tensioner node. Note how all lines of force intersect in a single point irrespective of assembly angles.



eliminating the thickness of the tile or any deviation in it from the tolerance stack.

CONTINUOUS CONTACT CONTROL FOR MICROCONTACT PRINTING USING PRECISION POSITIONING STAGE WITH OPTICAL FEEDBACK

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varying the position of each flexural bearing, compensating for errors in the stamp, roll, and substrate.

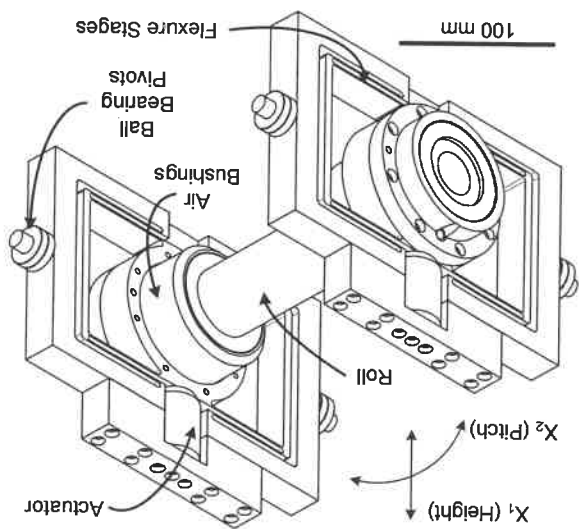


Figure 1. Schematic of the roll positioning stages, showing the roll supported by air bushings in vertical flexural stages. The stages are free to pivot in ball bearing mounts, allowing manipulation of both the roll height and pitch.

Each flexural bearing is fitted with full Wheatstone bridge strain gage arrays and passive Lorentz coils for full state feedback (position and velocity, respectively). Lorentz coil actuators, driven by analog current amplifiers, provide control effort at each flexural bearing (Fig. 2).

A quartz prism is mounted on a linear motion stage underneath the roll; this prism provides both a precision ground 'substrate' surface and enables real-time visualization of the stamp contact pattern. Two cameras

INTRODUCTION

This paper outlines the development of a precision positioning stage for experimental study of roll-based microcontact printing. In our current research, we are examining manufacturing scale-up of microcontact printing [1] for large area, high speed surface patterning. By mounting the microcontact printing stamp on a roll, we can achieve patterning rates characteristic of web-based processing while creating features on the scale of 100 nm. Stamp contact fidelity is critical for pattern transfer [2, 3]; we wish to examine the effect of contact force, pressure, and area on the contact pattern. In particular, we are interested in improving process precision through continuous process output feedback control.

To this end, we have constructed a precision positioning stage for modulation of the stamp-substrate interface in a roll-based printing configuration (Fig. 1). Design choices were made to facilitate continuous monitoring of the stamp-substrate contact pattern. In the following sections, we discuss the machine design, system characterization, position feedback control design, and outer loop feedback control of stamp contact using camera imagery.

DESIGN

Prior designs of microcontact printing machines have effectively utilized flexural bearings for precise roll positioning [4]. Our experimental printing machine employs two degrees of freedom: a precision-ground printing roll is supported by two air bearings, each in turn supported by vertical flexural bearings mounted in pivots (Fig. 1). By

are mounted in the machine to provide imagery of the contact pattern (Figs. 3 & 4).

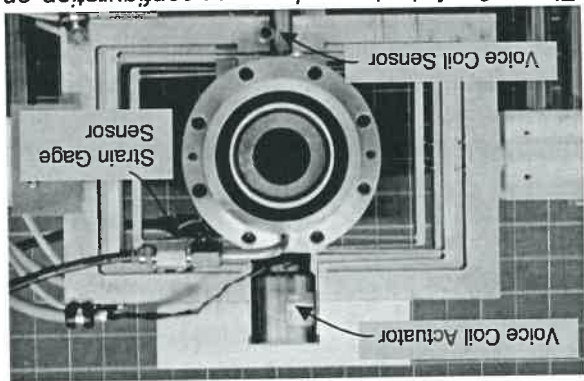


Figure 2. Actuator and sensor configuration on each roll bearing support stage (opposite stage symmetric).



Figure 3. The prism and cameras beneath the roll permit monitoring of the stamp contact region.

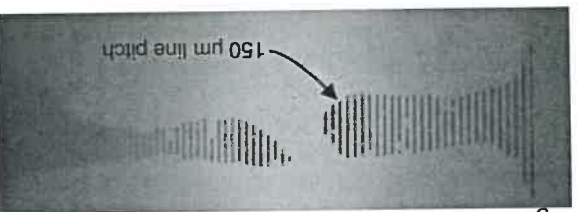


Figure 4. Camera images used to visualize failure modes and sensitivities. For example, defect modes can easily be detected: non-uniform contact (upper) and pattern collapse (lower).

CHARACTERIZATION

The position and velocity sensor gains were calibrated through static measurements with a micrometer and frequency response correlation, respectively. The resolution of each sensor (without low pass filtering) was extracted from the power spectral density of each signal (Table 1).

Sensor	Gain	Resolution
Strain Gage 1	0.262 mm/V	693 μ V
Strain Gage 2	0.308 mm/V	604 μ V
EMF 1	12.2 mm/V-s	732 μ V
EMF 2	19.6 mm/V-s	336 μ V
		6.59 μ m/s
		8.93 μ m/s

Table 1. Sensor gains and resolution.

The structural loop stiffness of the machine (through the bearings, roll, ground, and actuator mounts) was measured to be 3.57 N/ μ m using a static loading curve. This stiffness is suitably high for precise observation of microcontact printing contact mechanics.

The roll and air bushings have a combined eccentricity of +/- 1.9 μ m, measured with capacitance probes along the working surface of the roll.

The flexural bearing dynamics were determined through system identification using experimental frequency response data. While the actuators and sensors are mounted on each stage, the height (X_1 in meters) and pitch (X_2 in radians) of the roll were used as measurement variables, while force (U_1) and torque (U_2) were used to actuate the system. A fourth order state space model was found to sufficiently describe the system for the bandwidths of interest, though higher order dynamics can clearly be seen in the frequency response data (Fig. 4).

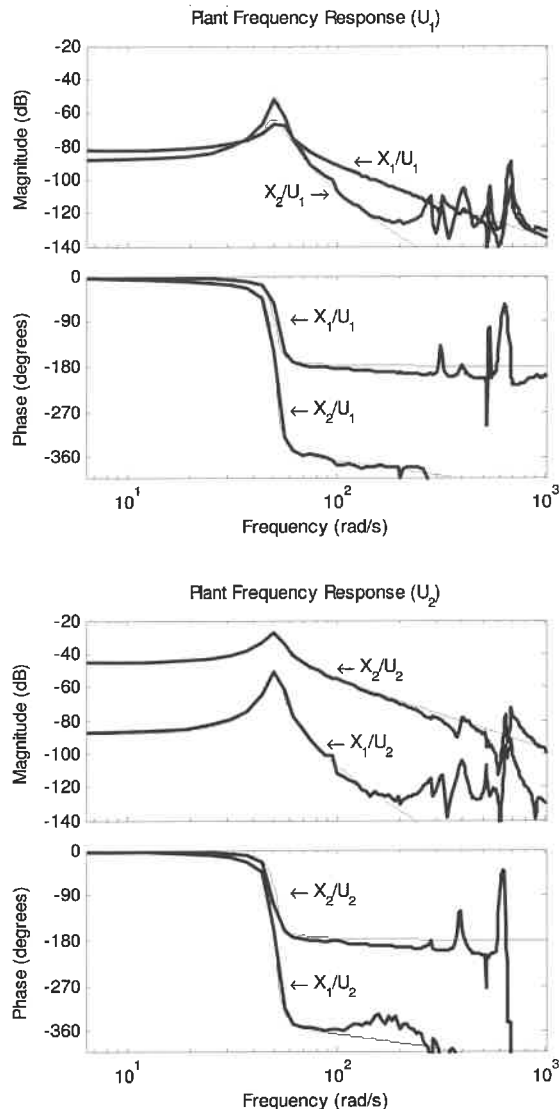


Figure 5. Plant frequency response illustrating low damping and coupling between system outputs. Fourth-order model (represented by thin line) fits primary dynamics of experimental response very well. U_1 and U_2 are force and torque; X_1 and X_2 are roll height and pitch.

POSITION CONTROL

Using the system model developed in the previous section, we use full state feedback control with integrators to modulate position. By using eigenvector assignment methods to place the closed-loop poles, the height (X_1) and pitch (X_2) modes can be separated, allowing independent treatment of these position variables.

The closed loop system has a -3 dB bandwidth of 30 Hz for both position variables, with settling times of 40 ms (Fig. 5). The closed loop position variables have resolutions of 72 nm and 0.50 μ rad.

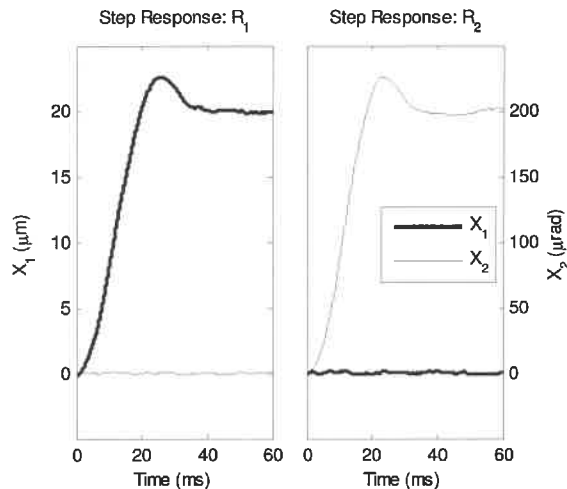


Figure 6. Closed loop step responses show fast response time and successful mode separation.

OUTER LOOP CONTACT CONTROL

Preliminary process investigations demonstrate stamp contact area and collapse are hyper-sensitive to stamp thickness variation. The two degrees of freedom in the positioning stage (height and pitch) permit compensating for errors in stamp parallelism or roll eccentricity. Using the cameras mounted in the machine, we have improved the process uniformity by implementing continuous control of the contact area under rolling conditions (ca. 1 mm/s) using optical feedback.

In the optical feedback loop, thresholding is applied the camera images and a fill factor extracted (Fig. 7). Prior knowledge of the camera frame size and the stamp pattern allow relating the frame fill factor to the stamp contact area. Desired roll height changes above each of the two cameras are transformed to changes in the roll positioning variables (X_1 and X_2).

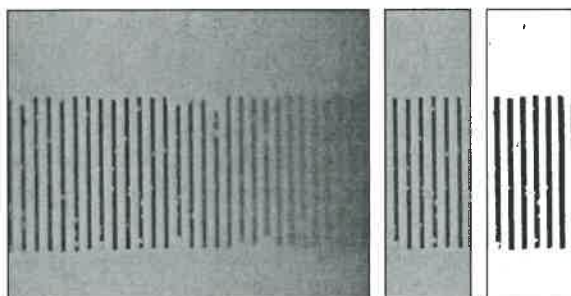


Figure 7. To extract a signal from contact imagery for feedback control, each camera frame (left) is cropped to the appropriate region of focus (center). This area is processed in real time to provide a thresholded image (right), where the percent fill factor is compared with a reference to compute the error signal.

The positioning stage bandwidth is higher than the camera frame rate (10 fps), permitting discrete integral control at 10 Hz while maintaining loop stability. The loop gain is set by considering a simple arc length model of the roll position / contact area transfer function. Despite contact area sensitivity to stamp thickness variation in the open loop case (constant roll position, Fig. 8), the closed loop system maintains uniform contact along the full length of the stamp (Fig. 9).

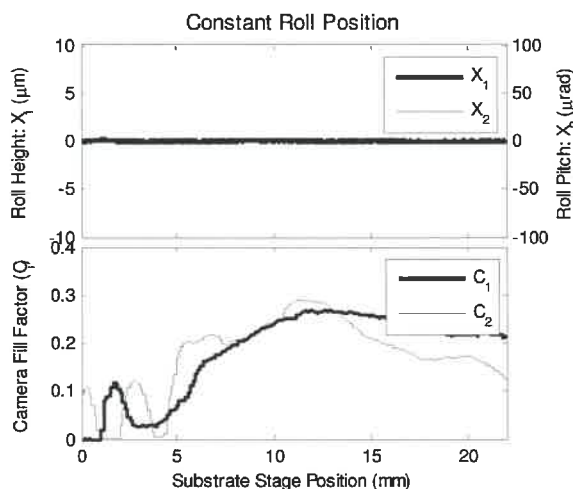


Figure 8. When the roll position (X_1 and X_2) is held constant, contact area (as measured by camera fill factors C_1 and C_2) is very sensitive to disturbances (primarily stamp thickness errors) as the roll is moved over the substrate. Rolling speed was ca. 1.25 mm/s; the horizontal axis represents substrate position under the roll.

CONCLUSION

A two degree-of-freedom roll manipulator has been designed and constructed for examining roll-based microcontact printing. The machine has high stiffness, small error motions, and submicron resolution through full state feedback control.

Cameras mounted in the machine permit in-process observation of the stamp contact region. We have demonstrated improved disturbance rejection in the rolling process by using contact images in a real time feedback loop.

This machine enables future studies of process performance, sensitivities, and control strategies.

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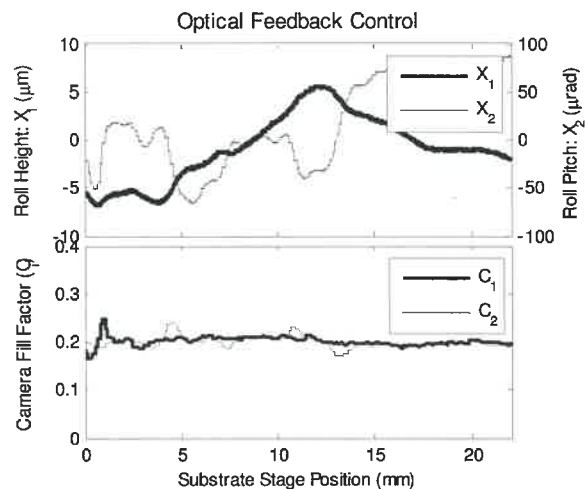


Figure 9. Closing a feedback loop around camera fill factors C_1 and C_2 rejects disturbances by manipulating roll position through X_1 and X_2 . This strategy results in a much more uniform contact area. Rolling speed was ca. 1.25 mm/s; the horizontal axis represents substrate position under the roll.

Progress on High Precision Maglev Stage Technology for Parallel High-Throughput Reflective Electron Beam Direct Write Lithography

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INTRODUCTION

REBL (Reflective Electron Beam Lithography) is a program for the development of a novel approach for high-throughput maskless lithography. The program at KLA-Tencor is funded under the DARPA Maskless Nanowriter Program. A DPG (digital pattern generator) chip containing over 1 million reflective pixels that can be individually turned on or off is used to project an electron beam pattern onto the wafer. The DARPA program is targeting 5 to 7 wafers per hour at the 45 nm node. This paper will describe a novel Maglev stage that enables this performance. The vacuum compatible rotary wafer stage has been developed in partnership with Philips.

THE NANOWRITER REBL CONCEPT

The contactless Maglev rotary stage architecture removes the throughput barriers associated with linear stage architectures where demanded aggressive stage acceleration limits throughput. The rotary stage will hold and expose 6 or more wafers simultaneously without turnaround overhead. The rotary stage operates quasi-static (also at high scanning velocities), with extremely low disturbing impacts on the writing process and with highly reproducible and predictable wafer- and stage motion.

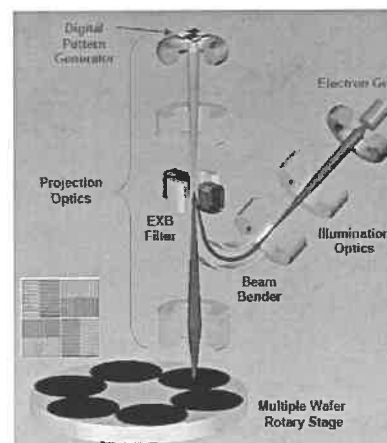


Figure 1: The nanowriter REBL concept

THE HIGH THROUGHPUT NANOWRITER MULTI COLUMN REBL CONCEPT

A single column will not deliver enough beam current at the needed blur to achieve high throughput and therefore a multiple column architecture with next generation smaller columns will be employed. The Rotary Stage platform offers a convenient system for multiple columns. As the number of columns increases many other system requirements are significantly reduced. The footprint of the machine is around 4 sq. meter.

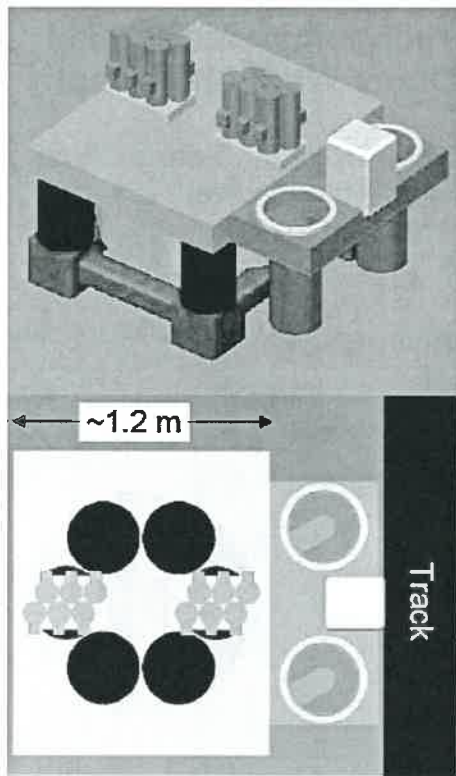


Figure 2: The high throughput system architecture

THE MAGLEV ROTARY STAGE

The rotary stage (mass=120 [kg], radius=1 [m]) consists of 9 actuators which generate the generalized forces to control the 6 degrees of freedom of the rotary stage. Eight of those actuators are hybrid electromagnets, a reluctance actuator (E-core shape) with a permanent magnet to achieve pretension or to compensate for gravity forces of the levitated rotor. In this way a frictionless and contactless active bearing is created that operates with almost zero dissipation. An ironless brushless DC motor is used to drive the stage in Rz direction. The position of the rotor with respect to the base is measured using IFM for x,y and multiple encoders for Rz. Inductive probes are used to accurately measure the magnetic gap needed for actuator linearization. Sensor and actuator redundancy is used to differentiate between and compensate for reproducible error motion and sensor or actuator artifacts. This involves systematic and novel (in situ and real-time recursive) calibration and control schemes.

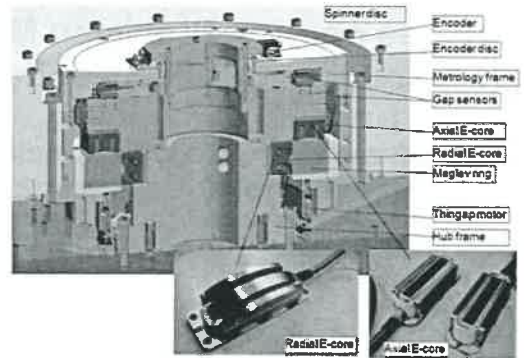


Figure 3: The Maglev active bearing and drive system

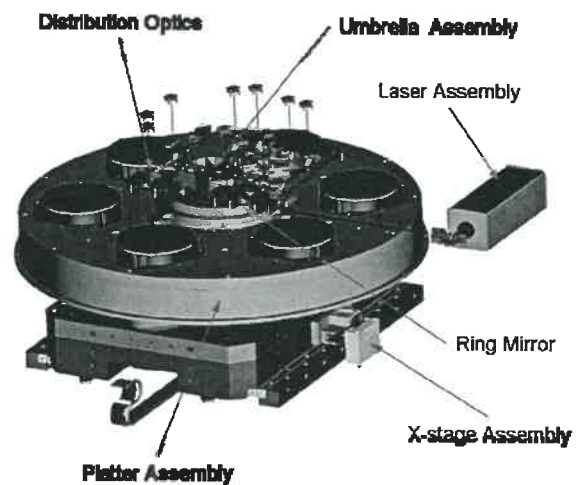


Figure 4: The Maglev rotary stage on a translating linear stage

STAGE PERFORMANCE

One of the main requirements of the stage besides being vacuum compatible and being magnetically stray field compliant and robust under operation, is the limited allowable (servo) error motion (for low frequencies, less than about 1 [μ m] and especially for frequencies above 100 [Hz], less than 1 [nm]) also for high scanning velocities (1-10 [Hz]). Besides this, the servo errors should be real time predictable in time and space up to 1 [nm] at all times during the writing process (22 nanometer node and beyond).

- **Hybrid electromagnet linearization:**
The parasitic stiffness is minimized from $2e5$ [N/m] to $1e3$ [N/m]. RMS Currents minimized 0.04 [A].
- **Balancing:** compensation in SW and HW demonstrated from $1e-1$ [kgm] up to $1e-3$ [kgm].

- **Unroundness:** within 1 [μm] and calibrated/compensated up to single digit [nm].
- **Alignment:** Relative position of principle axis of rotor with respect to metrology ring and maglev actuator surface are within $20\text{e-}6$ [m], $100\text{e-}6$ [Rad] in all directions and calibrated and compensated up to $10\text{e-}9$ [m].
- **Errors (without IFM):** measured at Rz velocity = 1 [Hz], above 100 [Hz] less than $10\text{e-}9$ [m], below 100 [Hz] less than $200\text{e-}9$ [m] in x and y direction and $0.5\text{e-}6$ [Rad] in Rx and Ry direction.

NOTE

This Project is supported by DARPA and KLA-Tencor under the DARPA Agreement # HR0011-07-9-0007

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Design and Experimental Characterization of a Large Range XYZ Parallel Kinematic Flexure Mechanism

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ABSTRACT

We present the design and experimental characterization of a novel parallel-kinematic flexure mechanism that provides highly decoupled and large motions along the X, Y, and Z translational directions. Motivated by applications in large-range multi-axis nanopositioning, the proposed design inherently accommodates the geometric constraints associated with integrating fixed-axis actuators. The motion characteristics of the proposed flexure mechanism are validated by means of finite element analysis as well as experiments. The design and fabrication of the hardware prototype and experimental setup are presented. Analytical and experimental results demonstrate a cross-axis error of less than 5.6%, actuator isolation less than 1.5%, and no perceptible motion direction stiffness variation over an XYZ motion range of 10 mm x 10 mm x 10 mm.

INTRODUCTION AND PRIOR ART

Flexure mechanisms derive motion from elastic deformation instead of employing traditional sliding or rolling interfaces. This joint-less construction entirely eliminates friction, wear, and backlash, resulting in highly repeatable motion. Further benefits include design simplicity, minimal assembly and maintenance, and potentially infinite life. These attributes allow flexure mechanisms to be widely employed in various design applications such as nanopositioning systems, scanners, precision alignment instruments, and MEMS devices.

In recent years, there has been a growing need for large-range multi-axis flexure mechanisms, driven by requirements in large-range nanopositioning. Nanopositioning systems are macro-scale mechatronic motion systems capable of nanometric precision, accuracy, and resolution [1], and are therefore vital to scanning probe-based microscopy, manipulation, and manufacturing [1-2]. However, most existing flexure-based, multi-axis nanopositioning systems are capable of approximately 100 μm motion range per axis. To broaden the impact of

scanning probe techniques in nanometrology and nanolithography, there is a need to increase this motion range by several folds [2-3].

Multi-axis motion capability in a flexure mechanism may be achieved via either a serial kinematic or a parallel kinematic configuration. While a serial kinematic design may be simply conceived by stacking one single-axis system on top of another until one achieves the desired degrees of freedom, this construction is often bulky. Moreover, moving cables and actuators adversely affect the precision and dynamic performance of the resulting motion system. Parallel kinematic designs, on the other hand, employ ground-mounted actuators and are often more compact. However, compared to serial kinematic designs, their main drawbacks include smaller motion range, potential for over-constraint, and greater error motions. Furthermore, parallel kinematic designs are non-obvious and difficult to generate.

Several macro-scale parallel-kinematic XYZ flexure mechanisms have been reported in the literature, but none provide large-range motion capability. A monolithic six-DoF compliant structure with a stroke of 100 μm per translational axis is reported in [4]. Another XYZ design, with 140 μm range per axis, is presented in [5]. In both cases, the motion range is restricted primarily due to inadequate geometric decoupling and/or actuator isolation between the multiple axes. A more detailed prior art discussion may be found in [6].

In this work, we present a novel XYZ parallel-kinematic flexure mechanism that meets the goal of large-range motion via the following attributes: I. The mechanism inherently provides geometric decoupling between the X, Y, and Z motion axes, allowing large un-constrained motions along each direction. II. It exhibits small error motions such as cross-axis errors and parasitic rotations, which are undesirable. III. It provides actuator isolation, thereby enabling the use of large-stroke, single-axis actuators.

PROPOSED DESIGN

The proposed design, shown in Fig.1, is based on a systematic and symmetric layout of two kinds of building blocks – rigid stages and parallelogram flexure modules (PFMs). The rigid stages are labeled as Ground, X stage, Y stage, Z stage, XY stage, YZ stage, ZX stage, and XYZ stage. This nomenclature is based on the primary mobility of any given stage – the X stage is constrained such that it has mobility along the x direction only, the XY stage is constrained such that it has mobility in the x and y directions only, the XYZ stage is constrained such that it has mobility in the x, y, and z directions, etc.

In addition to the rigid stages, there are 12 PFMs, grouped by color – green, blue, and red, which serve as motion constraints. The green PFMs (G1 through G4) deform primarily in the x direction and remain stiff in all other directions; the red PFMs (R1 through R4) deform primarily in the y direction and remain stiff in all other directions; and, the blue PFMs (B1 through B4) deform primarily in the z direction and remain stiff in all other directions.

As a consequence of this unique constraint arrangement, the X stage is constrained to move primarily along the x direction guided by PFM G1. Because of the high stiffness of the blue and red PFMs in the x direction, the x displacement of this X stage is passed on to the XY, XZ, and the XYZ stages. At the same time, the XY and XYZ stages remain free to move in the y direction because of the compliance of the red PFMs, and the XZ and XYZ stages remain free to move in the z direction because of the compliance of the blue PFMs.

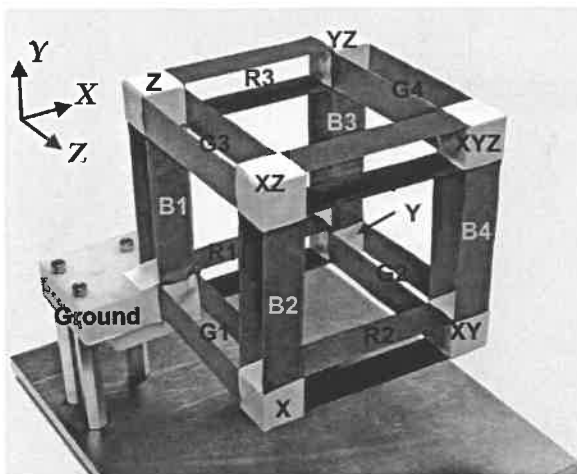


FIG 1. Proposed XYZ Flexure Mechanism Design

Given the symmetry of this design, similar motion behavior is exhibited in the other two directions as well. The y motion of the Y stage is transmitted to the XYZ stage without affecting the latter's mobility in the x and z directions. And, the z motion of the Z stage is transmitted to the XYZ stage without affecting the latter's mobility in the x and y directions. Moreover, the X, Y, and Z stages provide ideal locations for integrating large-stroke, fixed-axis x, y, and z direction actuators, respectively.

PROTOTYPE DESIGN AND FABRICATION

An experimental setup, shown in Fig. 2, was designed and fabricated to validate the qualitatively predicted motion performance of the proposed flexure mechanism. A non-linear static finite element analysis (FEA) was conducted using ANSYS, using BEAM4 and MPC184 elements. This analysis provided the basis for the overall size, detailed dimensions, and material selection of the hardware prototype.

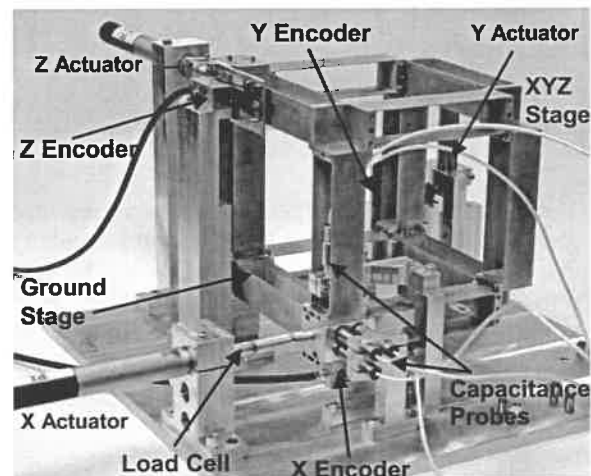


FIG 2. Experimental Setup

The proposed design comprises PFMs arranged in a fashion that makes monolithic fabrication very difficult, if not impossible. Therefore, the flexure mechanism is assembled from individual PFM modules with rigid mounting plates at each end. The PFM modules were equipped with rigid plates with appropriate kinematic features to achieve accurate and repeatable alignment during assembly, without the use of an external jig. As shown in Fig. 3, three rigid plates from three PFM modules come together at each corner of the flexure mechanism; the three plates are exactly constrained with respect to each other by means of dowels and mating slots. The dowel slots prevent the assembly from being over-constrained, and accurately

align the PFMs as they are tightened together with screws. To acquire the high tolerances necessary for alignment, the PFMs were cut from a precision-ground plate using wire EDM. The same dowel and screw features enable precise mounting of external sensors and actuators to the flexure mechanism. A precision-ground plate with dowel features aligns the actuator and sensor mounts to the mechanism.

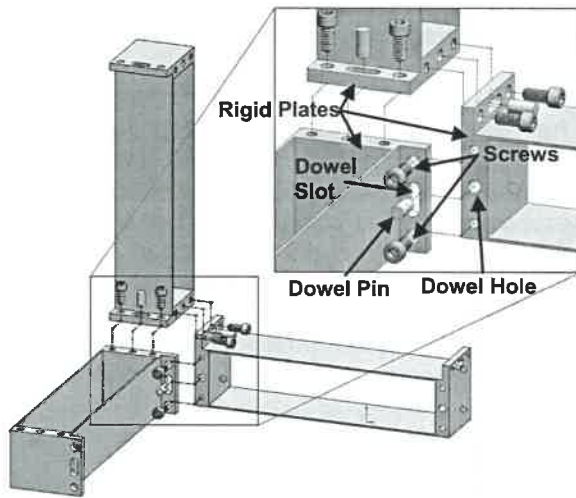


FIG 3. Flexure Mechanism Assembly with Alignment Features

The bearing characterization requires measurements of the absolute translational and rotational displacements at the input motion stages (X, Y, and Z stage) and the output motion stage (XYZ stage). High-resolution linear optical encoders are well-suited for measuring long range single-axis translations of the X, Y, and Z stages. Given the symmetry of the design, the error motions associated with an input stage are measured at the X stage only. Since these error motions are relatively small ($\leq 150 \mu\text{m}$), an arrangement of five capacitance probes is used to measure the y and z translations and the three rotations at the X stage. Similarly, the cross-axis errors and parasitic rotations at the XYZ stage are also measured using multiple capacitance probes. The force applied at the X stage is measured using a load cell. The X, Y, and Z stages are actuated over their entire motion range ($\pm 5 \text{ mm}$) using micrometers.

TEST RESULTS AND DISCUSSION

The following figures illustrate representative FEA results and experimental measurements over an XYZ motion range of $10 \text{ mm} \times 10 \text{ mm} \times 10 \text{ mm}$. They demonstrate the geometric decoupling between the three axes, the resulting

large, unconstrained motion range, and the small error motions. These figures use the nomenclature of $U_{\text{direction}}^{\text{STAGE}}$ to denote the translational motion of a particular stage (e.g. U_x^{XYZ} represents the x direction displacement of the XYZ stage) and $F_{\text{direction}}^{\text{STAGE}}$ to denote forces. Blue circles represent the experimental data with dashed fit lines, and red squares represent FEA predictions with solid fit lines.

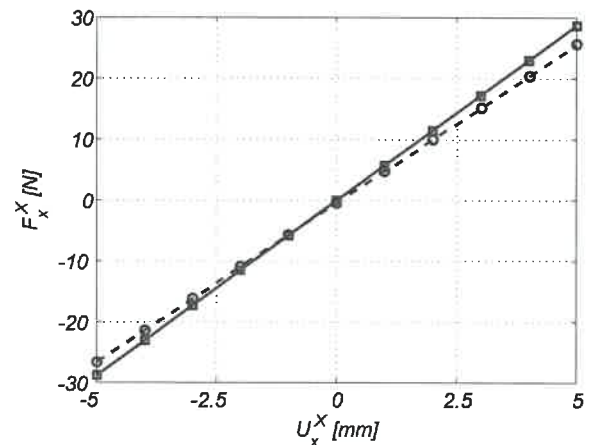


FIG 4. X Direction Force-Displacement Relation

Fig. 4 shows the stiffness associated with the x direction displacement of the X stage in response to an x direction force at this stage. This x direction stiffness is measured while the Y stage is held at -5 mm and the Z stage is held at $+5 \text{ mm}$. Additional FEA and experiments (not shown in Fig. 4) revealed minimal variance in the above x direction stiffness for all possible combinations of Y and Z stage displacements, confirming the highly decoupled motion. The force-displacement curve is linear, as expected, with a maximum force of 27 N at $\pm 5 \text{ mm}$.

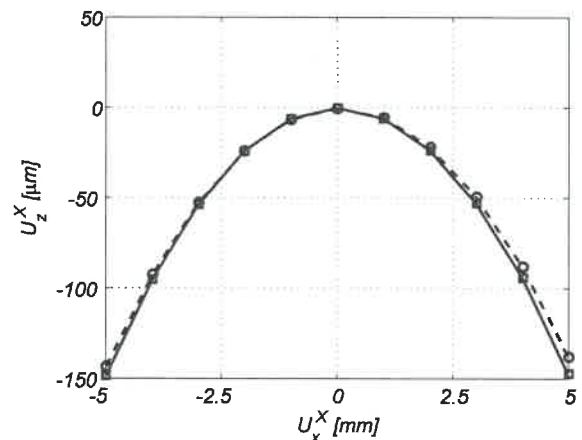


FIG 5. X Stage Error Motion (Z Direction)

Minimizing the translational error motions of the X stage (i.e. y and z direction error motions) is

important to accommodate large-stroke, fixed-axis actuators. Fig. 5 presents the error motion of the X stage in the z direction over the range of motion of the X stage while both the Y and Z stages are held at 0 mm. This structural nonlinearity is caused by the arc-length conservation of the green PFMs as they deform. The FEA model accurately predicts this motion with less than 3% error. This measurement was repeated for various combinations of Y and Z stage displacements, and agreed with the FEA.

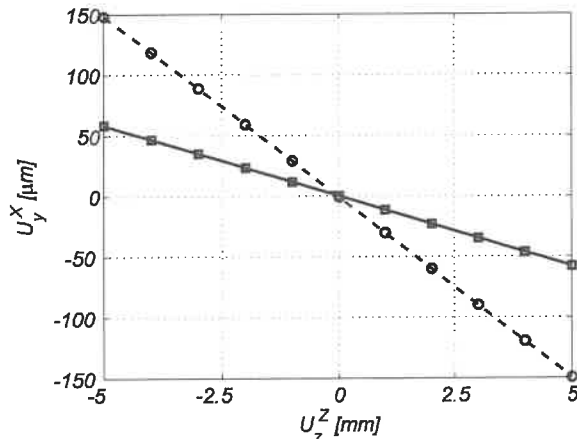


FIG 6. X Stage Error Motion (Y Direction)

Fig. 6 presents the error motion of the X stage in the y direction over the range of motion of the Z stage while both the X and Y stages are held at 0 mm in their primary directions. This error motion is mainly caused by the actuation of the Z stage in the z direction. This actuation of the Z stage creates a moment about the ground stage that loads the green PFMs (particularly G1 and G3) in the y direction. These green PFMs have a large, but finite stiffness in the y direction, and therefore deform by a small amount, resulting in a small y displacement at the X stage. The FEA model predicts this trend, but experimental data reveals that the mechanism is actually more compliant. This discrepancy is likely due to the FEA model assumption of a monolithic construction and fails to account for the large, yet finite joint stiffness associated with the interface of three PFMs at each corner of the assembled flexure mechanism.

Cross-axis error is the translational motion of the XYZ stage in a direction other than the intended input. Fig. 7 shows the z direction motion of the XYZ stage over the range of motion of the X stage while both the Y and Z stages are held at 0 mm in their primary directions. As with the X stage error motion, there is a kinematic-

quadratic component that arises from the arc-length conservation of the green PFMs. There is also a linear component that arises due to elastic and elastokinematic effects in the red and green PFMs. Again, the FEA model predicts this trend, but the experimental data shows a greater amount of cross-axis error than predicted. Other than the finite joint stiffness ignored in the FEA model, this discrepancy could also be due to the choice of BEAM4 elements, which might be inadequate in capturing all nonlinear effects.

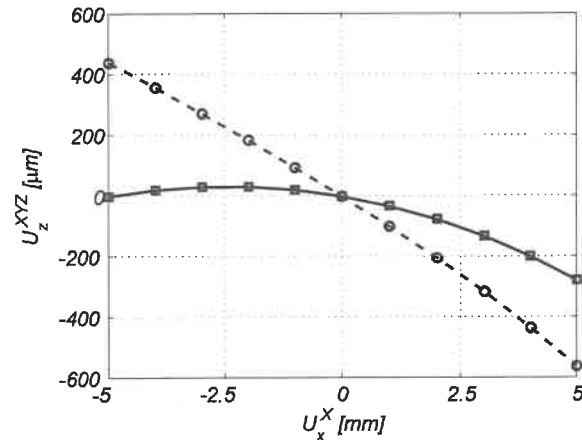


FIG 7. XYZ Stage Cross-Axis Error Motion

Ongoing research includes more comprehensive analytical and experimental characterizations, with a focus on understanding the discrepancies observed.

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DESIGN AND CHARACTERIZATION OF A VISUAL METROLOGY INSTRUMENT WITH A 5-AXIS MOTION SYSTEM

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KEYWORDS

Metrology, NIF, fusion target, instrument design, error budget, system characterization

ABSTRACT

This paper describes the design and characterization of a visual metrology instrument with a 5-axis motion system for the National Ignition Facility (NIF). Adhering to the principle of determinism, we predicted the level of accuracy and repeatability expected from the machine. This allowed us to meet the performance requirements and provide the end-users with a better understanding of the limitations of the machine.

The Non-Ignition Target Inspection Machine (NITIM, pronounced NYE-TIM), is a metrology system for characterizing non-ignition targets for the NIF at LLNL and for the Omega Laser Facility at the Laboratory for Laser Energetics. These targets are used in a wide range of laser driven experiments in Inertial Confinement Fusion (ICF), High Energy Density Physics (HEDP), basic science (e.g. astrophysics and materials science), and the development of diagnostics for those experiments. The diversity of roles of non-ignition targets leads to a multitude of target geometries. In order to fully inspect the wide variety of target sizes, shapes, and complexities, the NITIM's 5-axis motion system consists of an XYZ stack of linear stages along with two rotary stages, θ and ϕ . The motion system moves the target relative to two stationary vision systems that are orthogonal to each other and aligned with the motion system axes. Each vision system has a fixed magnification leg that serves as a reference, a

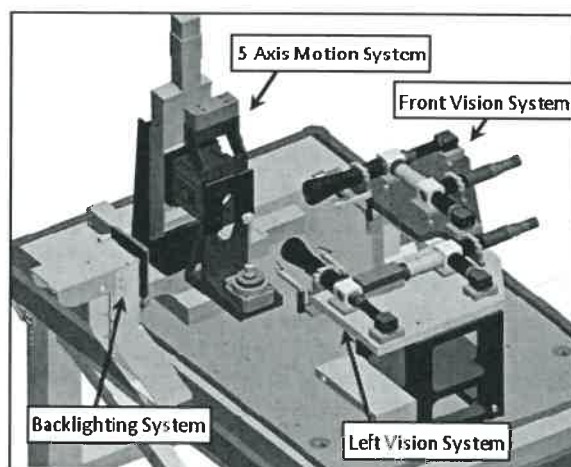


FIGURE 1. Isometric view of NITIM CAD model.

variable magnification (zoom) leg, digital cameras, and an eyepiece.

In order to meet the system requirements, various precision engineering design tools were used. Error budgets that included alignment errors and stage stiffness were used in the form of homogeneous transformation matrices (HTMs), and lists of error components decomposed into translations and rotations about the X-, Y-, and Z-axes of the global coordinate system. Various combinatorial rules were applied to the error terms to gain insight into and manage the sources of errors, and to predict the final accuracy of the motion system. Lumped mass modal analyses were performed to understand the contribution of each of the 5 motion stages to the predicted dynamic behavior of the system, and to help in the selection of stages.

This paper will also detail the methods used for characterizing the performance of the as-built system and discuss the discrepancies between predicted values and measured values. We will discuss the measurement of the motion system errors using a laser interferometer and laser autocollimator, characterization of the alignment of the stages, and the characterization of the accuracy of the system as a metrology instrument based on commercial and custom artifacts. We also compare predicted and measured dynamic performance of the system.

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Design and Testing of a High Accuracy Robotic Single-cell Manipulator

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INTRODUCTION

We have designed, built and tested a high accuracy robotic single-cell manipulator to be able to pick individual cells from array of microwells, 50 μm each. Design efforts have been made for higher accuracy, higher throughput, and compactness.

This proposed system will be used to help select and isolate an individual hybridoma from polyclonal mixture of cells producing various types of antibodies. It is important to be able to do this cell-retrieval task since a single isolated hybridoma cell produces monoclonal antibody that only recognizes specific antigens, and this monoclonal antibody can be used to develop cures and treatments for many diseases.

Our research's development of accurate and dedicated mechatronics solution will contribute to more rapid and reliable investigation of cell properties. Such analysis techniques will act as catalyst for quicker discovery of treatments and vaccines on a wide range of diseases including HIV infection, tuberculosis, hepatitis C, and malaria with potential impact on the society.

DESIGN OF SINGLE-CELL MANIPULATOR

The proposed system is designed to have a T-bar mechanism with two linear stages for XY-plane positioning to have higher stiffness and less structurally-inherent error. Figure 1 shows the design of the proposed system, *robotic single-cell manipulator*.

Precision is especially required in Z-axis movement for successful cell-retrieval procedure. Having independent structure for Z-axis motion on the XY positioner which provides large stiffness, the proposed cell-picker is designed to minimize error. Among many options considered for Z-axis motion, a rotational mechanism with a

voice coil actuator is selected because this results in relatively smaller reaction on the system and has advantages of direct drive.

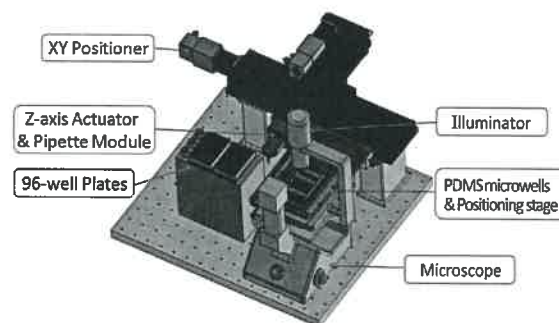


FIGURE 1. Conceptual design of the robotic single-cell manipulator.

PROTOTYPE OF THE PROPOSED SYSTEM

An old hard disk drive system is utilized to obtain Z-axis motion since it has a well-constructed electromagnetic motor and also well-built-in bearing. Arm extension, shown in Figure 2, has also been designed, constructed, and assembled to meet the design requirements of the system. Since the voice coil motor has a relatively small torque constant of 0.4 Nm/A, it is important to counter-balance the arm.

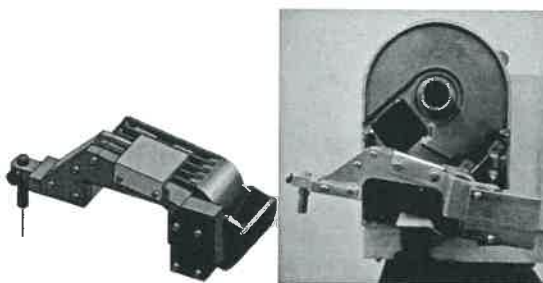


FIGURE 2. Rotational z-axis mechanism with the voice coil actuator and the arm extension.

The prototype, illustrated below in Figure 3, integrates the Z-axis and XY stage motion, real-time microscopy imaging, and cell manipulation with a NI PXI-controller centered as a main real-time controller.

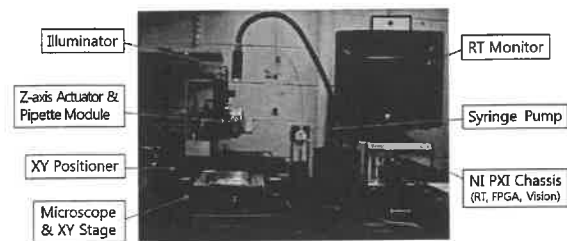


FIGURE 3. Physical configurations of the prototype of the robotic single-cell manipulator.

An inverse microscope is also self-designed and constructed assembling a charge-coupled device (CCD) camera and a zoom lens specified enough to observe a 10 μm cell or microbead, and a microscope stage with the resolution of submicron. In addition, a syringe pump, together with a glass pipette, is used to aspirate and dispose individual cells out of Polydimethylsiloxane (PDMS) microwells and in 96 well plates with the control of appropriate liquid amount and speed.

The overall scheme for the prototype system is illustrated in Figure 4.

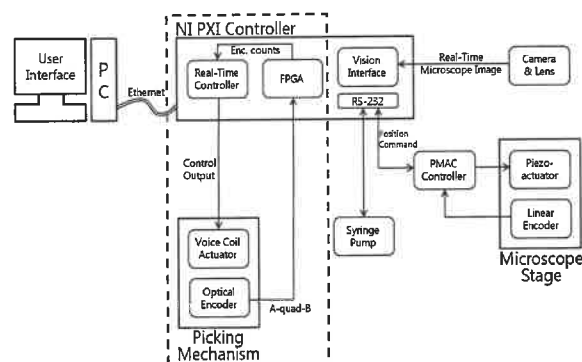


FIGURE 4. System schematic for the prototype of the robotic single-cell manipulator.

First, real-time microscope images of the PDMS microwells are transferred through the vision interface installed in the NI PXI Chassis, and those images are used to locate an individual cell or target of interest among a myriad number of microwells. The XY microscope stage is then used to move the PDMS microwells so as to

align the well of interest with the expected location of the end-effector, which is a glass micropipette driven fluidically by a syringe pump. After locating the target, the glass pipette mounted at the end of Z-axis actuator arm is lowered so that the tip touches the rim of the target well. To precisely control the tip position, an optical encoder is used as the actuator arm position sensor. A Field Programmable Gate Array (FPGA) is programmed to count the edges of the encoder A-quad-B output. Controlled by the main controller through serial communication, the proper amount of liquid is then aspirated by the syringe pump in order to pick the target cell or microbead out of the microwell. All of this process is shown through the user interface window for operators to observe.

CONTROL OF THE PROTOTYPE

For the prototype system, the XY microscope stage moves the PDMS microwells and commands for this movement are given from the main controller to the Universal Motion and Automation Controller (UMAC) through a serial communication as indicated in Figure 4. Linear encoders implemented on the XY stage provide a resolution of 0.16 μm , which is one edge count, together with an interpolator contained in the UMAC controller. This encoder position data is used for a simple PID control implemented on UMAC for a fast and smooth motion. Figure 5 shows a step response of the position-controlled XY stage.

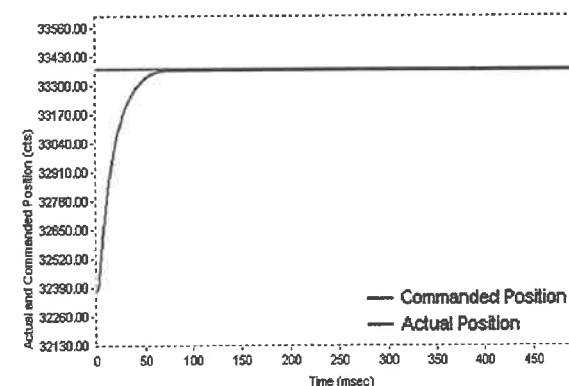


FIGURE 5. Step response of the XY stage with a PID controller.

It is also possible to lower the rise time for faster movement by increasing the proportional gain; however, this can easily yield large overshoot and slow settling, which are not what we are looking for in this application.

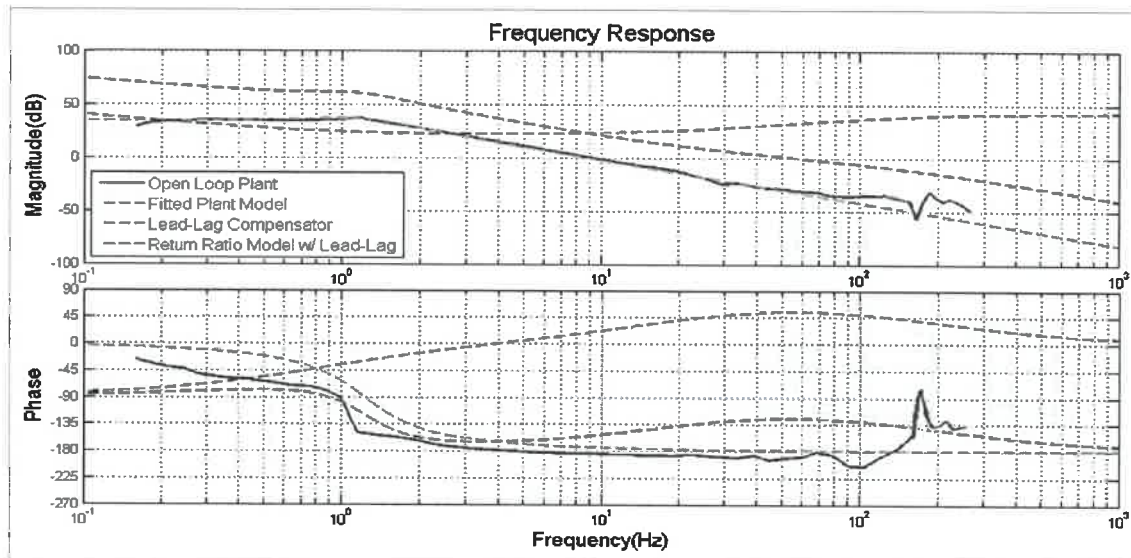


FIGURE 6. Frequency responses of the plant, compensator, and loop transfer function of the Z-axis mechanism with the controller.

As for the control of Z-axis motion, a loop shaping method is used. First, the frequency response of the open loop plant, which is the voice coil actuator, is measured and fitted with a mathematical model. A proper compensator is then designed for desired system characteristics such as a phase margin, gain margin, and crossover frequency. Since the plant has its first mode at around 150 Hz as shown in Figure 6, the crossover frequency is designed to be located at 60 Hz with the phase margin of 55°. The plot in Figure 6 illustrates measured frequency response of the plant and the fitted model together with the compensator's frequency response and the loop transfer function of the closed loop system. A step response of this closed loop system is shown in Figure 7.

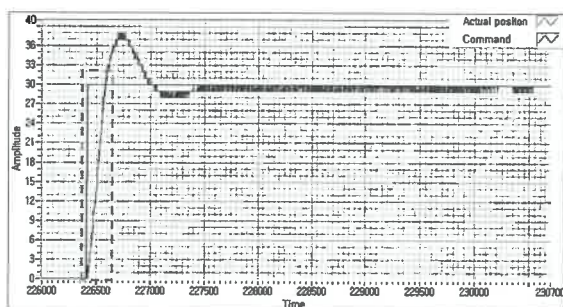


FIGURE 7. Measured step response of the closed loop system of the Z-axis actuator.

One tick of the time in Figure 7 indicates the control loop time of 50 μ s, and the amplitude is in the unit of an encoder count, which is equivalent to shaft's rotational displacement of 3 μ rad and the linear motion of the end-effector, 0.45 μ m. While this system has a short rise time of 5.5 ms, it also shows a 20% overshoot. This requires us to generate a trajectory command smooth enough to avoid the control output saturation and large overshoot. We implement a cubic polynomial for our smooth trajectory which is illustrated in Figure 8, and the measured trajectory response of the Z-axis mechanism is shown in Figure 9.

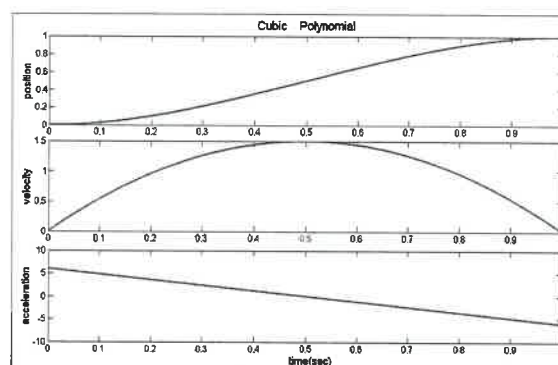


FIGURE 8. Position, velocity, acceleration profile of the cubic polynomial.

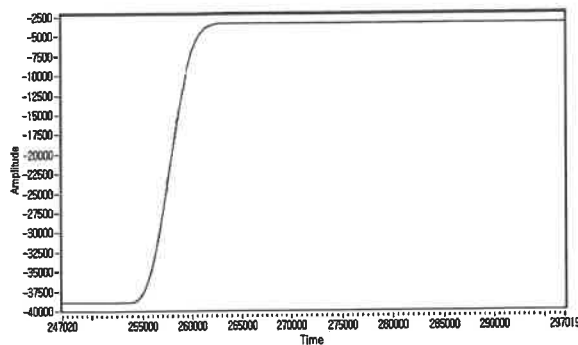


FIGURE 9. Measured trajectory response of the Z-axis mechanism; units of time and amplitude are 50 μ s and 3 μ rad, respectively.

As expected, the trajectory generation provides smooth motion to the Z-axis actuator without the control output saturation and overshoot.

RESULTS AND CONCLUSIONS

This prototype is built to test performances of the proposed system in terms of single-cell retrieval. This cell-retrieval is experimentally tested with the equivalent size, 10 μ m, of microbeads. Figure 10 below shows one block of PDMS microwells containing a number of 10 μ m microbeads; the left figure is before the cell-retrieval and the right one is after.

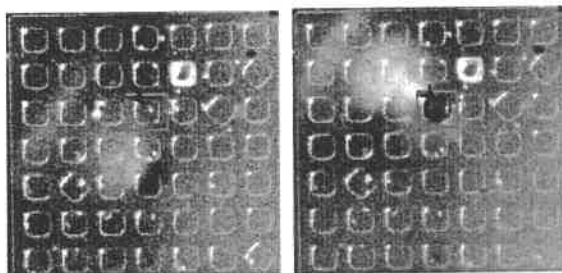


FIGURE 10. Monochrome CCD images of PDMS microwells before (left) and after (right) the cell-retrieval with the prototype.

The small white dot pointed with a block arrow indicates the picked microbead. The retrieval process itself takes average of 5 to 10 seconds depending on how long microbeads have stayed in microwells. The testing of the prototype results in reliable cell-retrieval with the success rate of more than 90 percent. More experiments with actual living mammalian cells will be conducted as a future work.

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Development of a 3D Surface Mapping System to Inspect Capsule Fill-Tube Assemblies used in Laser-Driven Fusion Targets

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INTRODUCTION

Lawrence Livermore National Laboratory is currently working on the National Ignition Campaign (NIC) with the goal of demonstrating inertial confinement fusion (ICF) [1] by generating a thermonuclear ignition and energy gain in a laboratory setting utilizing the National Ignition Facility (NIF) [2].

Figure 1 shows a schematic of the ICF target used in NIC experiments (for a complete description the reader is referred to [3] and [4]). Positioned in the center of the hohlraum is a capsule fill-tube assembly (CFTA) containing the deuterium-tritium (DT) fuel. The CFTA consists of a hollow Ge- or Si-doped plastic fuel capsule ranging in diameter between 2.2 mm and 2.6 mm and an attached 150 μm diameter glass-core fill-tube that tapers down to a 10 μm diameter at the capsule.

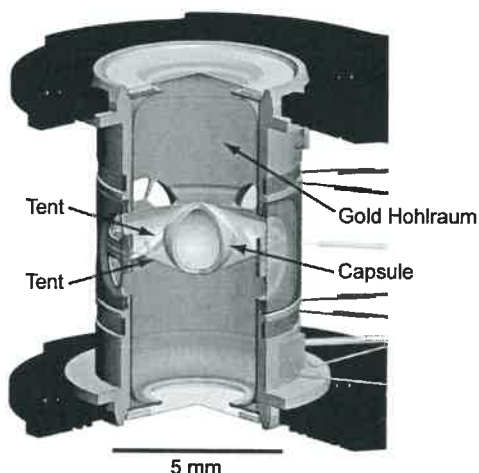


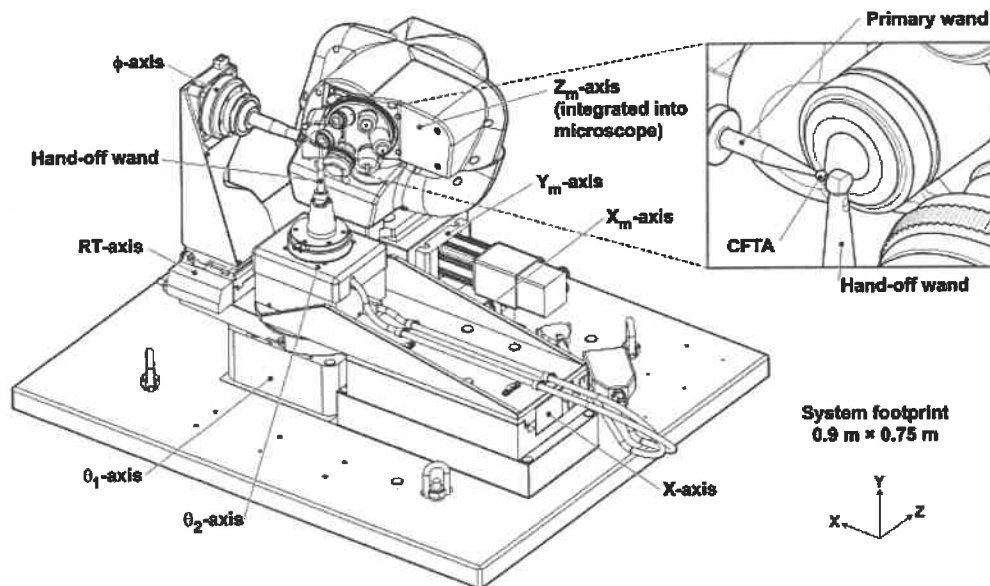
Figure 1: Schematic of the ICF target used in NIC experiments.

The development of a 3D surface mapping system used to measure the surface of a CFTA is presented. The mapping system is an enabling technology to facilitate a quality assurance program and to archive 3D surface information of each CFTA used in fusion ignition experiments. The 3D Surface Mapping System is designed to locate and quantify surface features as small as 50 nm in height and 300 nm in width. Additionally, the system will be calibrated such that the 3D measured surface is related to the capsule angular coordinate system to within 0.25 degree (1σ), which corresponds to approximately 5 μm linear error on the capsule surface.

3D SURFACE MAPPING SYSTEM

The 3D Surface Mapping System, shown in Figure 2 and Figure 3, comprises seven linear and three rotational axes that are used to manipulate the Capsule Fill-Tube Assembly (CFTA) to achieve a deterministic 4π steradian inspection using a surface profiling confocal microscope. To reduce operator error and input effort, the system is fully automated. Automation includes manipulation of the capsule for low magnification image acquisition, data upload, and low magnification image analysis that generates defect coordinates which are in turn used to image at high magnification, providing quantitative height information on a subset of relevant defects.

The primary fixture for handling the CFTA is a vacuum wand mounted to the ϕ -axis stage. The fill-tube is inserted into the inner diameter of the conical tip such that the fill-tube is captured within the wand (basic functionality of the wand is described by Montesanti et.al. [5]). A second



θ and ϕ is about the y-axis and x-axis, respectively
Figure 2: Line diagram of the 3D Mapping System.

vacuum wand, the hand-off wand, will provide the capability of exposing the area initially covered by the primary wand.

MEASUREMENT PRINCIPLE

To obtain a 3D surface map of the capsule, a surface profiling confocal microscope is used. The confocal microscope provides a set of individual images that together form a complete bright field surface map of the capsule. This surface map is performed at a low magnification to reduce measurement time. Higher magnification objectives are then used to investigate individual surface features at a higher fidelity, providing quantitative height information. To achieve this, the ϕ -axis and the θ_1 -axis are rotated such that approximately 90% of the capsule surface is covered. Figure 4 shows the geometric coverage pattern at a magnification of 50x ($NA = 0.55$) producing 460 individual patches in approximately 2.5 hours.

To obtain the remaining surface area, the capsule is transferred from the primary wand to the hand-off wand exposing the area that was previously covered. In order to image the exposed area, the fill-tube is bent utilizing the θ_1 -axis, θ_2 -axis and RT-axis, see Figure 5. At this point, the remaining surface area is scanned by moving only the microscope (*i.e.* the capsule is held stationary).

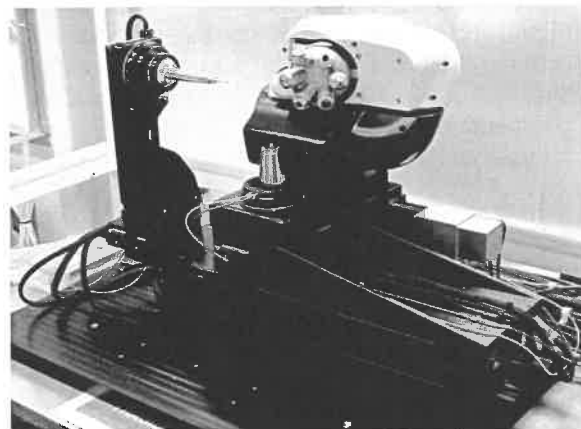


Figure 3: Photograph of the 3D Mapping System with the primary wand mounted.

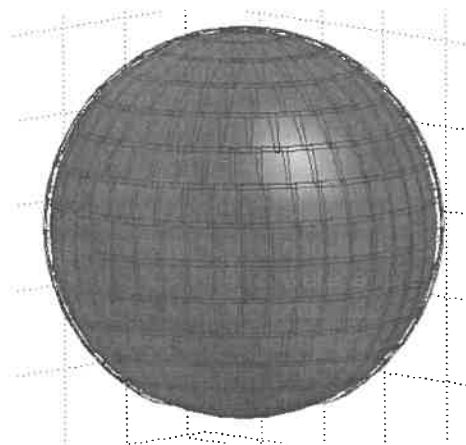


Figure 4: Imaging principle used to perform the measurements of a capsule covering approximately 90% of the surface area.

MEASUREMENT COMPARISONS BETWEEN OPTICAL AND MECHANICAL EDGES FOR A SILICON MICROMACHINED DIMENSIONAL CALIBRATION STANDARD

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ABSTRACT

A mesoscale dimensional artifact based on silicon bulk micromachining fabrication has been developed and manufactured with the intention of evaluating the artifact both on a high precision coordinate measuring machine (CMM) and video-probe based measuring systems. This hybrid artifact has features that can be located by both a touch probe and a video probe system. A key feature is that the physical edge can be located using a touch probe CMM, and this same physical edge can also be located using a video probe.

While video-probe based systems are commonly used to inspect mesoscale mechanical components, a video-probe system's certified accuracy is generally much worse than its repeatability. To solve this problem, an artifact has been developed which can be calibrated using a commercially available high-accuracy tactile system and then be used to calibrate typical production vision-based measurement systems. This allows for error mapping to a higher degree of accuracy than is possible with a typical chrome-on-glass reference artifact.

Details of the designed features and manufacturing process of the hybrid dimensional artifact are given, and a comparison of the designed features to the measured features of the manufactured artifact is presented and discussed.

Measurement results are presented using a meter-scale CMM with submicron measurement uncertainty; an optical CMM with submicron measurement uncertainty; a micro-CMM with submicron measurement uncertainty using three different probes; and a form contour instrument.

INTRODUCTION

Optical CMMs have submicron resolution, but it is difficult to certify their measurement uncertainties at the submicron level. This is due both to problems with specific illumination with optical CMMs [1] and the accuracy of the calibration artifacts used to calibrate optical CMMs [2].

The silicon micromachined artifact design and manufacturing methods were presented in [3], [4], [5]. Feedback from users suggested that micro-CMMs would also benefit from this type of calibration artifact. A second design iteration for fabrication with 150 mm wafers was made. The result of the second design iteration is shown in Figure 1.

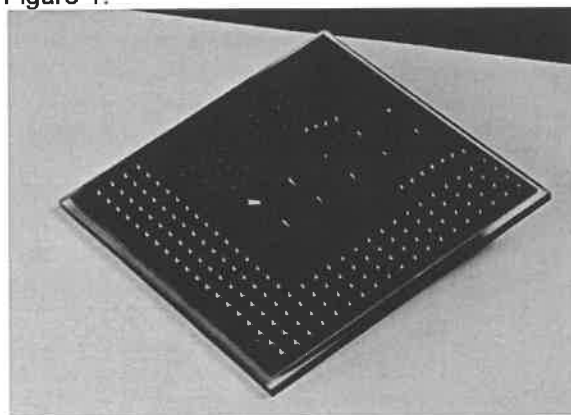


Figure 1. This design was created to operate in a micro-CMM, from [6]

Preliminary results for measurement results of this artifact have been published in [6]. These results compare measurements using a meter scale CMM (Leitz PMM-C Infinity¹, workspace is

¹ Certain commercial equipment, instruments, or materials are identified in this paper in order to adequately describe the experimental procedure. Such identification does not

1.2 m × 1.0 m × 0.6 m) with a capability for expanded measurement uncertainty (coverage factor $k=2$) of $(0.34 + 1.2L) \mu\text{m}$ (L in m), and an optical CMM (OGP Avant, maximum permissible error (MPE) in 2-D of $(1.2+2L) \mu\text{m}$ (L in m)). As the comparison with the OGP Avant has previously been published, that comparison is not included in this extended abstract.

The artifact was then evaluated on a high accuracy optical CMM (Mitutoyo Ultra Quick Vision, with a 1-D MPE in 1-D of $(0.25+L/1000) \mu\text{m}$, L in mm), a micro CMM (Mitutoyo M-NanoCoord) with a stated MPE of $(0.3+L/1000) \mu\text{m}$, L in mm, and a form/contour measuring machine (Mitutoyo Formtracer CS-3100, MPE of $(1+10L/1000) \mu\text{m}$, L in mm).

MEASUREMENT METHODS

All measurements are made based on the setup illustrated in Figure 2. The top surface ($\langle 100 \rangle$ silicon plane) and the West edges of the west-most cavities were used to establish a Cartesian coordinate system, and the midpoint of the South edge of cavity 1 was used to establish the origin. In order to eliminate possible bias in bidirectional measurements, we only evaluated results for South edges, and the pitch distance between South edges and the reference cavity (South edge of cavity 1).

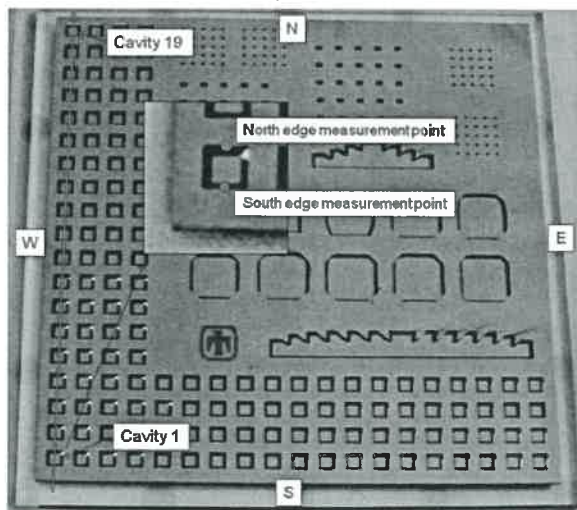


Figure 2. The reference location is at Cavity 1; measurements are made on S and N edges. Expanded measurement uncertainties, comprising an uncertainty and coverage factor for confidence, are calculated per the ISO guide

imply recommendation or endorsement by the authors, Sandia National Laboratories, Micro-Encoder Inc., or Mitutoyo America. The reported measurements are for informational purposes only.

[7],[8]. Common practice is to provide coverage factor for 95% confidence (typically $k=2$).

Meter-scale CMM

At the time the measurements on the meter-scale CMM were made, its capability was not accredited. An evaluation of this capability estimates that for 3-D measurements, the best capability at this laboratory is $\pm(0.34+1.2L) \mu\text{m}$ at a 95% confidence (L in meter) [9]. Because the measurements with the meter-scale CMM were performed at $20.02^\circ\text{C} \pm 0.05^\circ\text{C}$, no temperature correction was performed on the data. Other components of measurement uncertainties are associated with the artifact itself. The determination of the edge locations are made by probing the top of the artifact, and the etched $\langle 111 \rangle$ sidewalls, as shown in Figure 3.

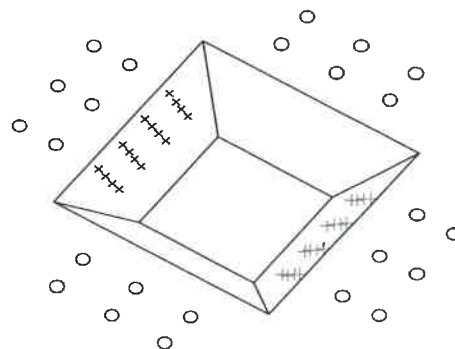


Figure 3. The top surface ($\langle 100 \rangle$ silicon) is evaluated as a plane from the points shown with the red dots. Sidewalls are also evaluated as planes. The edge is constructed by intersecting the $\langle 111 \rangle$ etched plane with the top plane.

In general, the $\langle 100 \rangle$ surfaces had very good reproducibility on the CMM (pooled standard deviation from each local plane of $0.034 \mu\text{m}$). The South $\langle 111 \rangle$ planes have poor surface finish (combined non-flatness and roughness as large as $1.2 \mu\text{m}$). Monte Carlo simulations of the effect of the non-flatness and roughness of the $\langle 111 \rangle$ walls on unidirectional measurements, including CMM capability, result in an expanded uncertainty ($k=2$) of $0.4 \mu\text{m}$ [6]. Based on experimental and simulation results, we use the larger of $(0.34+1.2L) \mu\text{m}$ or $0.4 \mu\text{m}$ as the measurement uncertainty for South edge distances for the meter-scale CMM.

High Accuracy Optical CMM

The high accuracy optical CMM was calibrated to meet its manufacturer specifications of $(0.25+L/1000) \mu\text{m}$, L in mm, when measuring aligned with a machine major axis.

Measurements were made at 22.7°C, then, corrected back to 20.0°C using $3.34 \times 10^{-6}/K$ as the coefficient of thermal expansion (CTE) for silicon. The uncertainty in temperature measurement is assumed to be $\pm 0.5^\circ C$ (rectangular distribution), and the uncertainty in the CTE is assumed to be $\pm 0.5 \text{ ppm}$ (rectangular). Assuming a rectangular distribution for the optical CMM specifications, we calculate a length dependent expanded uncertainty for the temperature correction of 0.82 ppm. Combining instrument and temperature correction terms, we obtain $(0.29 + 1.4L) \mu m$. Based on the simulation results of [6] for the uncertainty due to the artifact, we use the larger of $(0.29 + 1.4L) \mu m$ and $0.4 \mu m$ as the measurement uncertainty for South edge distances using the optical CMM. Figure 4 shows a photograph of a typical edge as seen by the optical CMM.

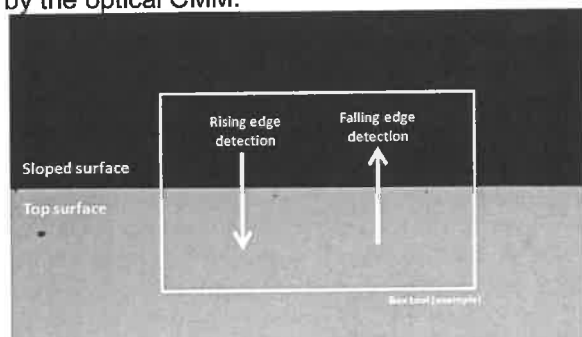


Figure 4. Two methods for edge detection.

Figure 5 compares edge detection methods on the same edge. These measurements were made on different days.

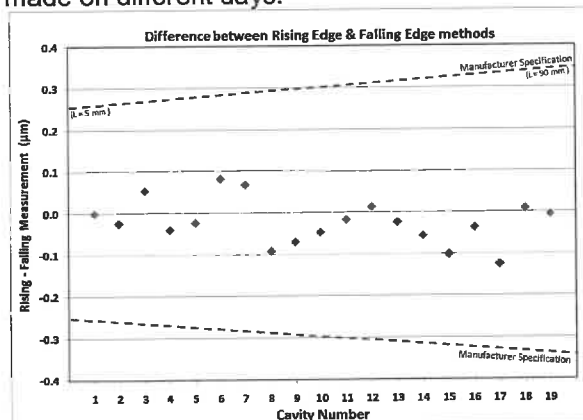


Figure 5. Comparison of edge distances when using rising edge vs falling edge detection.

Figure 6 shows the repeatability (mounting the artifact with clay on the optical CMM table) of 100 measurements. The standard deviation of the measurements is 28 nm for the edge location plotted (south edge of Cavity 1 to south

edge of Cavity 19). Uni-directional and Bi-directional repeatability using other cavities were also measured and showed similar repeatability.

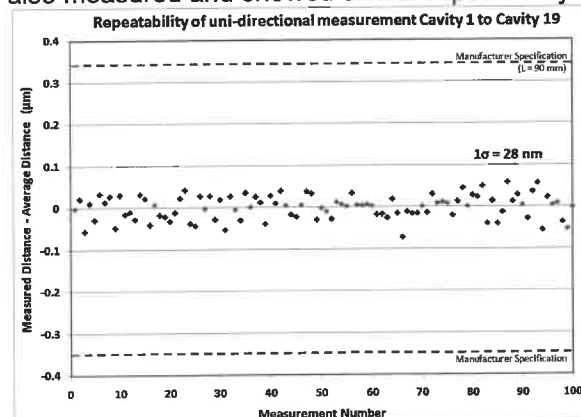


Figure 6. Repeatability of 100 measurements, using clay to fix the artifact on the optical CMM. The standard deviation was 28 nm.

Micro CMM

The Micro CMM (M-NanoCoord) has a specified MPE of $(0.3 + L/1000) \mu m$, L in mm. Measurements were made at 20°C, so no temperature corrections were made. Two different M-NanoCoords were used. One was equipped with both an ultrasonic touch probe (UMAP) and an optical vision probe; the other, with a long-range ultralow force scanning probe (LNP). We similarly estimate expanded measurement uncertainty to be the larger of $0.4 \mu m$ or $(0.35 + 1.2L) \mu m$, with any of the three probes. A comparison of the measurements is shown in Figure 7.

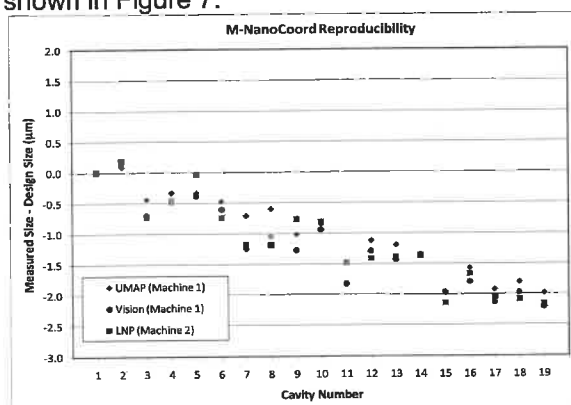


Figure 7. M-NanoCoord measurement reproducibility with three probes (UMAP, vision, and LNP) and two machines. Measurements are compared to artifact design size.

The intercomparison of 3 probing systems and two machines show excellent reproducibility, with a pooled standard deviation (3 probing systems, 18 cavities) on the order of 110 nm.

Contour Instrument

The contour instrument (Formtracer CS-3100) uses a stylus, and traces along the path with the long arrow shown in Figure 2. Figure 8 shows a typical setup for the contour instrument.



Figure 8. Setup for measurement of Formtracer. The stylus would trace points on the top and sidewalls. Intersection points were calculated, and distances calculated from the intersection points.

Repeated measurements using the Formtracer had a $0.2\ \mu\text{m}$ standard deviation. This is well within the machine specification for MPE of $(1+10L)\ \mu\text{m}$.

Intercomparison of Measurements

We compared results from each of the four measurement instruments by plotting design size (5 mm steps) minus measured size at each measurement location. The three M-NanoCoord data measurements are averaged.

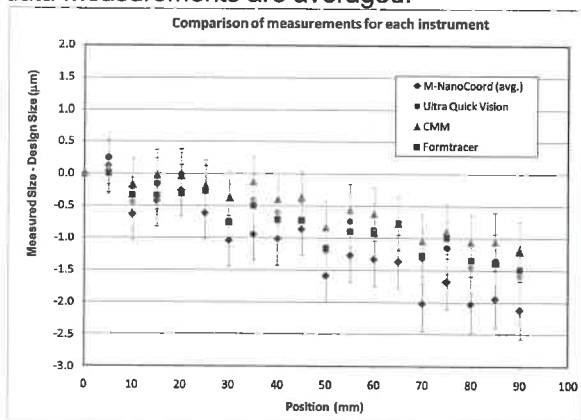


Figure 9. Comparison of measurements for each of the measurement instruments. (CMM is the meter scale CMM). Bars indicate estimated expanded measurement uncertainty ($k=2$). The Formtracer uncertainty is not shown, as it is significantly larger than the other instruments.

There is very good correlation between the four different instruments. All measurements agree within about $1\ \mu\text{m}$ or less. When considering measurement uncertainties, the measurements

overlap. This indicates that this artifact could potentially be used to calibrate optical CMMs and hybrid sensor CMMs. Since we did not give detailed instructions for setting up the measurement, we believe that much of the variation may be attributed to setup differences. It is interesting to note that the meter-scale CMM measurements trace to gage blocks calibrated at PTB, while the M-NanoCoord measurements trace to NMIJ, and the Ultra Quick Vision traces to NIST and NMIJ.

All instruments showed repeatability much better than the instrument MPE specifications.

ACKNOWLEDGMENTS

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Mitutoyo Corporation of Japan contributed the M-NanoCoord measurements.

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A SIX-DEGREE-OF-FREEDOM MOTION CHARACTERIZATION OF BALLSCREW-DRIVEN STAGE USING A SINGLE UNIT OF AN OPTICAL ENCOER

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INSTRUCTIONS

A high precision multi-degree-of-freedom (DOF) motion measurement technology is important for industrial activities such as manufacturing and inspection because those physical quantities such as the linear and angular displacement are key parameters for keeping and improving quality control of products [1-3]. The laser interferometer calibration system is the most famous instrument for the measurement of displacement. Also, the laser autocollimator can measure the pitch and yaw error, but cannot detect the roll error, which is defined as the angular displacement about the normal axis of the target. Also, those are high-end compared with that of an optical encoder. Many works have been done to measure motion errors. Saito and Kimura *et al.* introduced an optical method for measurement of three angular motion errors by using three pieces of position sensitive detector (PSD) and developed the compact new micro angle sensor [1]. Kim *et al.* developed a six-DOF measurement system using a circular grating target, but the small measurement range is limited within the diverging beam spot size applying on the circular grating target [2]. Also, Kim *et al.* reported a novel compact measurement system for measuring a single linear displacement and two angular motions in a nanostage, but it is not able to measure the roll motion [3].

Many studies have been mainly focused on measurement of only motion errors of the stage, but they were restricted to measure not only motion errors but also displacement with a single unit of an optical sensor, simultaneously. Also, the relation between motion errors and dynamic characteristics such as frequency response and driving mechanism of the stage was not investigated.

In this paper, the measurement method for six-DOF motion error of the stage was presented, and its optical sensor was constructed and its performance was evaluated.

MEASUREMENT

A simple measurement method was proposed, which is able to measure five motion errors along the driving axis in real time. The actuator was chosen as ballscrew-driven stage as shown in Fig. 1. The overall configuration of the optical sensor is shown in Fig. 2. The linear scale is mounted onto the moving stage and the stage travels along the x direction.

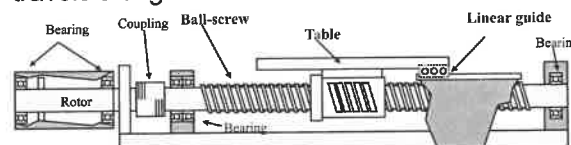


FIGURE 1. Configuration of ballscrew-driven stage.

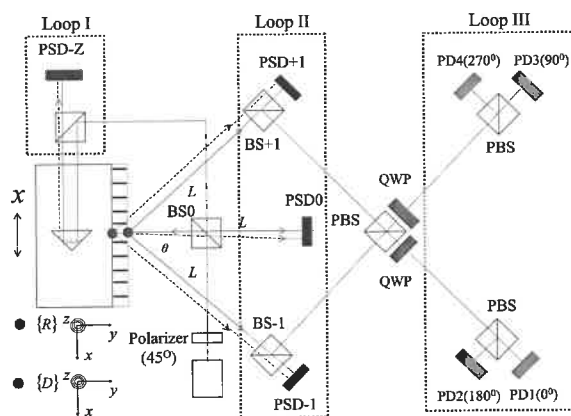


FIGURE 2. Configuration of measurement system.

The optical sensor consists of three loops. In Loop I, Y axis displacement error can be measured and in Loop II four-DOF motion error measurement is performed. A stabilized He-Ne laser ($\lambda = 632.8 \text{ nm}$) is used as an optical source with controlled intensity and frequency, and a polarizer is placed after the laser for linear polarization (45°). Through the beam splitter (BS0), the laser beam is reflected and it is projected onto the linear grating diffracted into each order ($+1^{\text{st}}$, 0^{th} , -1^{st}), respectively. The 0^{th} diffracted beam incidents back to BS1 and on PSD0 and the $\pm 1^{\text{st}}$ diffracted beams are divided

into two beams by BS2 and BS3 and projected onto PSD+1 and PSD-1, respectively, as shown in Fig. 2, where, u , v and y_z are the deviated displacement from the origin of PSD and the subscripts indicate the diffraction order from the grating. L is the distance from the beam spot position of the grating to the center of PSD, and θ is the diffraction angle.

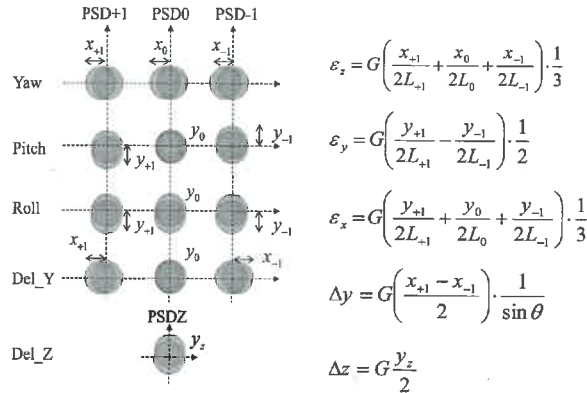


FIGURE 3. Beam position change according to errors.

In Loop III, the moving displacement along the x direction is measured by using the circular polarization interferometer. Two sinusoidal signals with 90° phase difference were measured and a perfect Lissajous figure was made with a resolution of 0.4nm noise level. Figure 4 shows the measurement result of motion error in ballscrew-driven stage with a stroke of 40mm. Ballscrew-driven stage is feedback-controlled with a laser interferometer. Through the calibration process, the resolution was 0.03 arcsec for an angular direction and 20nm for linear direction, respectively. The encoder performance was tested with a laser interferometer (Agilent). As shown Fig. 5, it was well agreed.

As shown in Fig. 6, six-DOF motion errors were successfully measured within a range of 40mm. It was thought that ballscrew and ball motion had a significant effect on pitch and roll error, which indicates that the stiffness at a yaw direction is bigger than that at other directions. Also, it can be seen that Z direction motion error was continuously increased, which indicates that the stage moved downhill.

CONCLUSION

A six-DOF motion error was successfully measured. The optical sensor showed a resolution of less than 0.03 arcsec. for angular, 20nm for linear errors, and 0.4nm for displacement along the traveling direction, respectively. It was confirmed that this encoder

unit is versatile to evaluate not only the motion but also characterization of stage construction. As a result, the proposed optical system is expected to be applied for production evaluation of a high precision stage.

Acknowledgements

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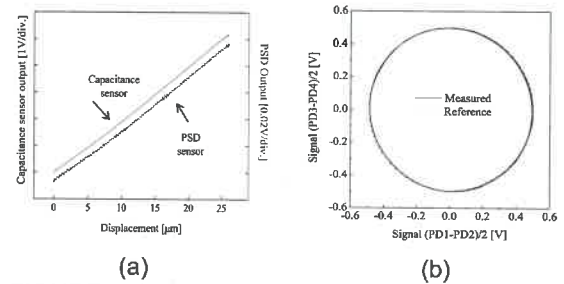


FIGURE 4. (a) PSD calibration, (b) Lissajous Figure.

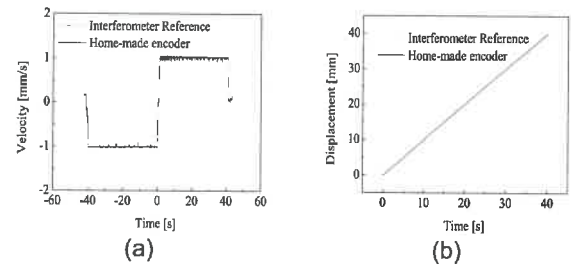


FIGURE 5. Encoder test : (a) Velocity, (b) Displacement.

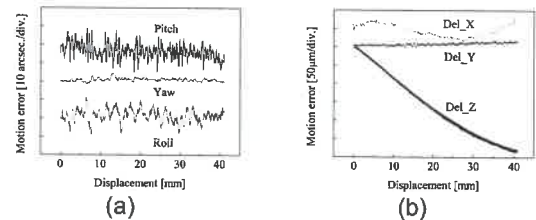


FIGURE 6. Measurement results of angular and linear motion errors.

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VERIFYING THE CAPABILITY OF A THREE AXIS VIBRATING MICRO-SCALE CMM PROBE TO OPERATE IN A NON-CONTACT MODE

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INTRODUCTION

Co-ordinate measuring machines (CMMs), capable of high accuracy tactile measurements of micro-scale features on millimetre-sized objects, have been available for many years. Such machines, commonly known as micro-CMMs, were initially developed by national measurement institutes [1] and subsequently by instrument manufacturers in response to demands by modern manufacturing engineers.

Existing high precision three-dimensional (3D) positioning stages, which are a constituent part of micro-CMMs, have proved more than adequate in operating to required accuracies [2]. However, the current state of probing technology limits the volumetric measurement uncertainty of micro-CMMs to approximately 100 nm [3]. Developments in the area of micro-co-ordinate metrology are now dependent on new probing systems being capable of reliable surface detection to a measurement uncertainty of several tens of nanometres.

Any new probe designs should address all the known issues that affect the accuracy of tactile probing on the micro-scale. Known issues of tactile probing include unwanted adhesion of the probe on the measurement surface, surface interaction forces causing false triggering of the probe and "snap-in" of the probe to the measurement surface, and plastic deformation of the surface under high contact forces during measurement [4].

An additional issue facing micro-CMM probe design is the lack of suitable styli whose effective working length with respect to their tip diameter (aspect ratio) is sufficient to contact currently produced high aspect ratio features. Also, it becomes more difficult to ascertain the size and form of the stylus tips for tip diameters less than 0.1 mm.

THE NPL VIBRATING MICRO-CMM PROBE

To address these issues, a vibrating micro CMM probe has been designed by the National Physical Laboratory. The probe consists of a triskelion (three-legged) MEMS device assembled with a sphere tipped micro-stylus [5], as shown in figure 1.

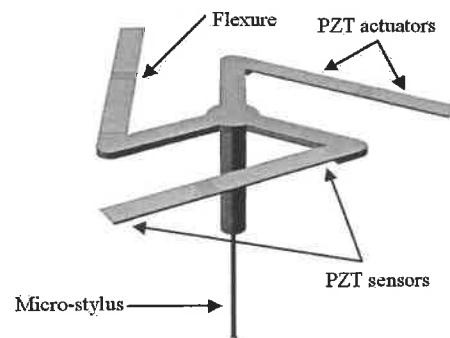


Figure 1. Schema of the NPL vibrating micro-CMM probe

The triskelion element is manufactured using typical silicon fabrication techniques, metal deposition and chemical etching. The flexures are made of nickel and a piezo-electric (PZT) film is deposited onto the flexures using a sol-gel spinning technique. The entire device is fabricated on a silicon wafer, but the active elements of the device are silicon free. Each flexure is 2 mm long, 200 μm wide and nominally 20 μm thick. The PZT elements are nominally 7 μm thick. The PZT film is used in the device for vibration actuation and sensing. The entire chip is 14 mm by 14 mm by 1 mm [5].

The sphere-tipped stylus is manufactured from tungsten by using a hybrid process of wire-electro-discharge-grinding and one-pulse-electro-discharge [6]. Previous investigations have shown that styli with tip diameters of 70 μm can be reproducibly fabricated using this method. The tip diameters vary by approximately 5 μm [7].

The micro-CMM probes are assembled using semi-automated desktop assembly, as described elsewhere [6]. Each completed micro-CMM probe is qualitatively tested to establish the orthogonality of the stylus and functionally tested by applying a lateral force on the stylus to verify sound adhesion between the assembled parts. A photograph of a fully assembled triskelion chip is shown in figure 2.

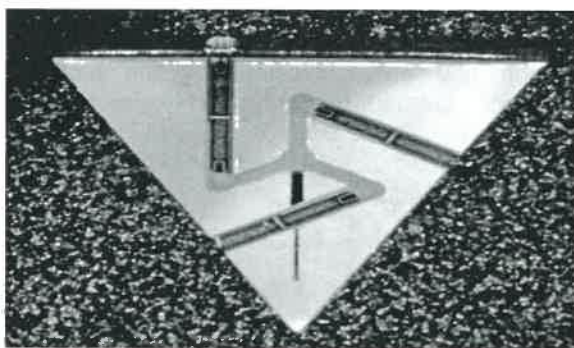


Figure 2. A completed prototype micro-CMM probe. The stylus shaft is 100 μm diameter, reducing to 50 μm with a 70 μm sphere tip.

Through a review of the surface interaction forces present during micro-scale probing [8], it was confirmed that the vibrations of this device would be sufficient to allow it to operate in a non-contact mode.

A similar method to counteract the surface interaction forces while performing co-ordinate measurements on the micro-scale has been pioneered by a standing wave probe [9], but the design discussed here has the advantage that it can vibrate in any axis or any direction so that the probing vector is always normal to the surface being measured.

VERIFICATION OF NON-CONTACT PROBING

To verify the ability of the probe to operate in a non-contact manner, a set of tests were undertaken that bring a surface (the gauging face of a tungsten carbide gauge block) into contact with the stylus tip. To fully investigate the non-contact probing characteristic of this probe, a repeatable triggering point, before contact is made, must be defined. To locate this trigger point, verification must include full contact with the reference surface so that all comparisons are made to a constant reference point.

The probe is held in a mount that includes all the required electrical connections. This mount is shown in figure 3.

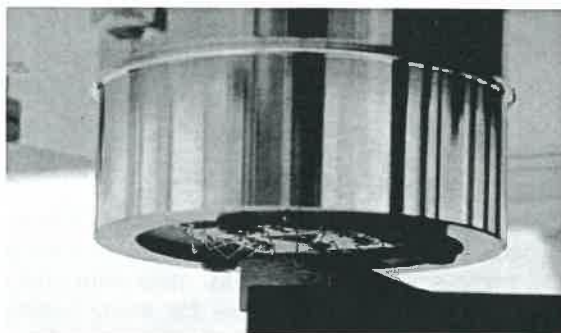


Figure 3. The probe in its mount, suspended above a tungsten carbide test surface. The probe is artificially highlighted in red.

This mount is attached to a kinematic stand, secured using a type II Kelvin clamp. The vibration of the probe is monitored from above using a Doppler laser vibrometer focused through a 20 $\times$ magnification objective. A photograph of the measurement setup is shown in figure 4.

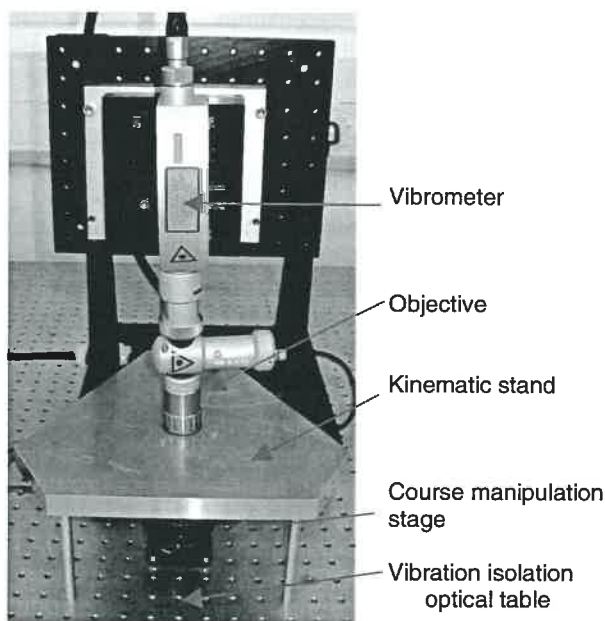


Figure 4. An image of the probe interaction setup.

The test surface (a tungsten carbide gauge block) was mounted on a piezo-electric nano-positioning stage. This stage has been calibrated over a 50 μm range using a coherence scanning interferometer. The piezo-

electric nano-positioning stage is mounted on a large range manual lab jack for coarse manipulation. This manipulation stage is shown in figure 5.

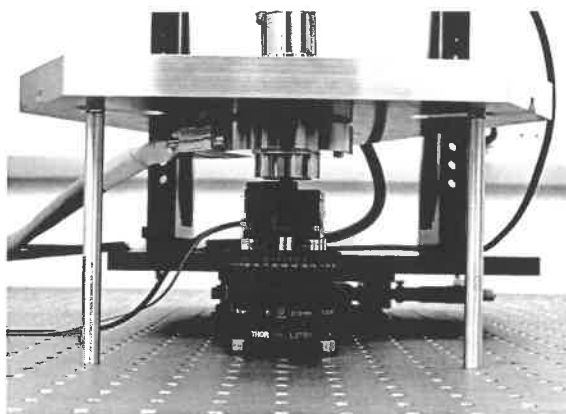


Figure 5. The manipulation stage below the micro-probe

RESULTS

The ratios between the changes in vibration amplitude and the approach distances of the reference surface were calculated. When there is no interaction between the probe and the surface, this ratio is nominally zero, and when the probe is making contact with the surface, this ratio is nominally one. Any intermediate ratios are the result of a non-contact interaction between the probe and the reference surface. The results of an interaction between the probe

and a measurement surface are shown in figure 6.

These results show an interaction occurring in the z direction, with the piezo-electric nano-positioning stage moving in $0.2 \mu\text{m}$ steps. The position of the reference surface was defined as the position whereby the ratio first passed unity.

DISCUSSION

The results, shown in figure 6, show that direct contact with the reference surface causes aberrations in the results of the ratio calculation. Several ratios are calculated to have values in excess of 1.25, which indicates that the vibration amplitude is changing more than the displacement of the reference surface would suggest. This anomaly could be caused by irregular vibrations occurring when the probe is in direct contact with the reference surface.

Several data points with ratios close to zero can be seen after zero distance. This is to be expected as when the reference surface is in full contact with the probe, any change in surface position will have no effect on the vibration amplitude, which is already at its minimum.

To further demonstrate the effect on the surface interaction forces on the vibration of the probe, a more detailed section of the graph is shown in figure 7.

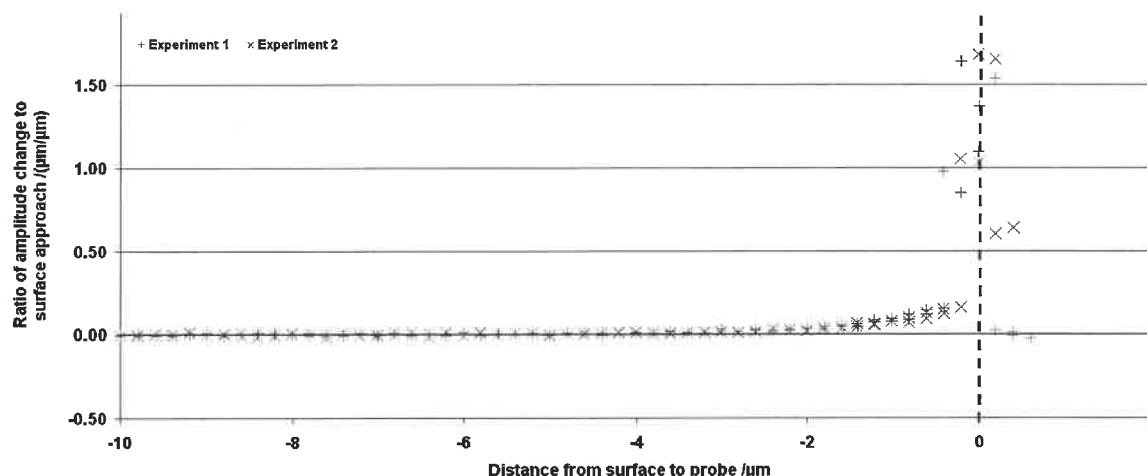


Figure 6. A plot of the ratio of amplitude change of the probe to the approach of the measurement surface. Deviations from the zero line indicate an interaction between the surface and the microprobe.

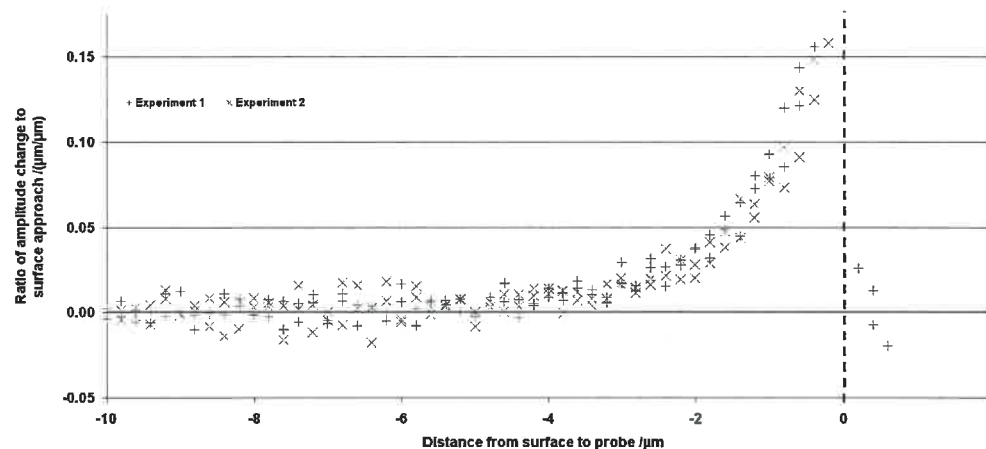


Figure 7. A plot of the ratio of amplitude change of the probe to the approach of the measurement surface. This graph shows only the surface interaction portion of the ratio range.

FUTURE WORK

It can be seen from figure 7 that the effect of the surface interaction forces results in a repeatable change in vibration amplitude. Although repeatability of the detection (and, therefore, counteraction) of surface interaction forces is required, a more essential requirement of the probe is repeatable surface triggering. This surface triggering can only be fully investigated if the position of the reference surface with respect to the probe is known to a higher accuracy than currently recorded, *i.e.* better than 200 nm. Therefore, experiments will continue with ever-smaller reference surface approach increments.

The repeatability of the piezoelectric nano-positioning stage over the period of each step is estimated as ± 15 nm. Experiments will be repeated using finer step sizes to ensure higher resolution data is obtained. However, the true position of the reference surface can only be confirmed within the ± 30 nm measurement uncertainty of the nano-positioning stage. The position of the measurement surface may be known to better than ± 45 nm. In depth analysis may improve performance, however, if this is not possible, a more accurate stage will need to be used.

A full set of verification tests will be completed, characterizing the vibration of the probe in 3D. To fully realize this probe as a working concept, the signals from the PZT sensors will be verified against the Doppler laser vibrometer to allow the probe to operate by self-sensing vibration changes. A rigorous uncertainty calculation will be developed to prove the accuracy of the probe. This uncertainty budget will include terms

for the calibration of the piezo-electric nano-positioning stage and its ability to repeatably position the measurement surface relative to the probe to the nanometre accuracy required. The final results will also report the isotropicity of the probe; a feature of modern CMM probes that assists in reducing measurement uncertainty.

ACKNOWLEDGEMENTS

This work was funded by the UK National Measurement Office Engineering and Flow Metrology Programme 2008 to 2011. The authors would like to thank Christopher Jones (NPL), Dong-Yea Sheu (NTUT) who developed the stylus production techniques [7], Arne Burisch (TU Braunschweig) who developed the assembly method [6] and the Microsystems and Nanotechnology Centre (Cranfield University) for producing the triskelion devices.

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ULTRA-PRECISION LINEAR MOTION METROLOGY OF A COMMERCIALLY AVAILABLE LINEAR TRANSLATION STAGE

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INTRODUCTION

Many new compact ultra-precision linear translation stages with exceptionally long ranges of motion on the order of tens of millimeters and positioning resolutions on the order of a nanometer are finding their way into emerging nanotechnologies. This class of positioning systems typically has specified linear error motions on the order of a few hundred nanometers or less and angular error motions on the order of $10''$ or less. However, measuring and certifying the positioning performance of these new stages with off-the-self instrumentation suggested by existing standards [1-3] can be very challenging if the test uncertainty ratio (TUR) is to be greater or equal to 4:1 as indicated in Ref. [4]. This is because sources of measurement uncertainty that were previously considered insignificant when measuring in the micrometer regime are very significant when measuring in the nanometer regime. In fact, this level of measurement is approaching the limitation of 5 parts in 10^8 for most off-the-shelf laser systems.

To address the challenges in characterizing this class of linear positioning system with appropriate TURs, we began an effort to extend our measurement capabilities for this operational regime and to measure the performance of a commercially available nanopositioner. This paper gives an overview of this effort. Descriptions of the measurement setups used to determine the six error motions of the stage are provided and each are followed by a list of the significant contributors to the combined measurement uncertainty of each positioning error measurement. The resulting measured positioning errors are then stated along with their associated TURs. Finally, the limitations in

our measurement capabilities are identified and a list of suggestions for enhancing the TURs for specified positioning parameters are provided.

NANOPOSITIONER

The system under evaluation is a single axis linear nanopositioner, Aerotech Inc. ANT130 Series [5]. The system is manufactured from Invar and has a travel range of 110 mm (± 55 mm) with a specified positioning resolution of 1 nm. The stage is driven by a linear motor and its position is determined by a linear encoder with a Zerodur scale. With the carriage at the home position (center), the stage has a length of 230 mm, a width of 130 mm, and a height of 45 mm.

LINEAR POSITIONING SET-UP

Linear positioning error [3], *EXX*, was measured using a commercially available single-pass displacement measuring interferometer. The heterodyne system, seen in Fig. 1, consists primarily of a beam splitter, two retroreflectors, and a Helium-Neon laser source with a vacuum wavelength accuracy of $\pm 0.1 \mu\text{m/m}$. Interferometer reference and measurement signals are captured by fiber optic couplings and transmitted via fiber optic cables to a measurement board where displacement is determined with a resolution of 0.62 nm. Each linear positioning (LP) measurement is corrected for changes in laser wavelength due to changes in refractive index of air and for stage encoder growth. When the measurement process is initiated, the refractive index is determined using a form of Edlen's equation [6,7] and by measuring temperature, barometric pressure, and relative humidity with a thermistor, pressure transducer, and humidity sensor with standard uncertainties of $\pm 0.002^\circ\text{C}$, ± 9 Pa, and $\pm 0.6\%$,

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respectively. Changes in refractive index are then tracked using a commercially available wavelength tracker that can detect a change in refractive index of 1.1×10^{-9} . An Invar fixture, designed to have the shortest possible thermal loop, supports and positions the interferometer optics with a dead path of approximately 1.6 mm. Alignment of the laser with the axis of travel is achieved using a combination of a laser position sensing detector (PSD) and a precision laser alignment fixture and is estimated to be within 100 ".

The measurement setup resides on top of a passive vibration isolation table located in a environmentally controlled laboratory where the temperature is controlled to $20.6^\circ\text{C} \pm 0.3^\circ\text{C}$. An enclosure encompassing the measurement setup is used to help minimize the effects of thermal fluctuations created by the laboratory ventilation system. Stage temperature is monitored with platinum resistance thermistors positioned on and around the nanopositioner. All measurements are controlled and monitored remotely from a location outside of the laboratory to further minimize disturbances to the environment.

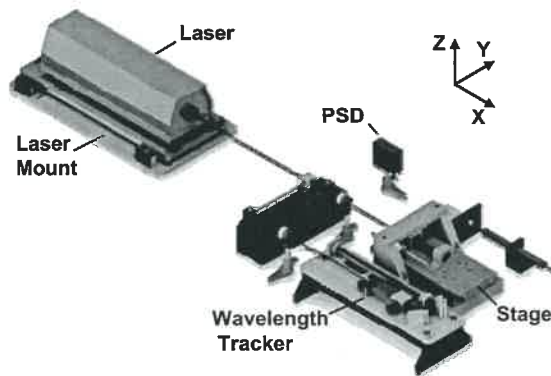


FIGURE 1. EXX measurement setup

LINEAR POSITIONING UNCERTAINTY

A Type B uncertainty analysis [8] based on the methods outlined in [9, 10] revealed that the combined standard uncertainty in measuring a point located at maximum stage travel (110 mm) is approximately 12 nm. The analysis also revealed that the combined standard uncertainties in bidirectional repeatability (R) and accuracy (A) for five bidirectional LP measurement runs are 16 nm and 19 nm, respectively. Assuming a coverage factor of $k=2$, R and A can be determined with

expanded uncertainties of approximately 32 nm and 38 nm, respectively. Many sources of measurement error contribute to the combined measurement uncertainties. The most significant contributors are listed in Table 1.

TABLE 1. LP uncertainty contributor estimates

environmental variation	u_{EVE} : 8.0 nm
misalignment	u_{MA} : 7.4 nm
wavelength accuracy	$u_{\lambda\lambda}$: 6.4 nm
refractive index of air	u_{Mn} : 3.7 nm
optics nonlinearity	u_{NL} : 2.9 nm

LP MEASUREMENT AND RESULTS

Many linear positioning tests, consisting of five 110 mm bidirectional runs of 23 random target positions within $\pm 30\%$ of a nominal 5 mm positioning interval, were performed to evaluate the linear positioning error, EXX . The measured data from the test having the lowest R value is provided in Fig. 2. The A and R values resulting from this test are $115 \text{ nm} \pm 38 \text{ nm}$ ($k=2$) and $47 \text{ nm} \pm 32 \text{ nm}$ ($k=2$), respectively.

STRAIGHTNESS & ANGULAR SETUP

The two straightness errors, EYX and EZX , and the three angular errors, EAX , EBX , and ECX were determined by measuring the lateral deviations of a moving straightness reference with linear displacement sensors [2], as seen in Fig. 3. A 203 mm x 26 mm x 26 mm mirror (straightness reference) with a flatness of 20 nm is mounted to the nanopositioner carriage via a kinematic coupling, which is used to accurately locate the straightness reference with respect to the stage encoder and displacement sensors. Three sets of three balls, which make up half the coupling, are adhered to the three nonmetallic sides of the straightness reference at its Airy points to minimize deflections. Employing three sets of balls provides the ability to accurately rotate and locate the reference between measurements which, in turn, helps to minimize the uncertainty in mapping the straightness errors of the reference, e.g., straightedge reversal and white light interferometry. The other half of the kinematic coupling is adhered to an aluminum plate bolted to the nanopositioner carriage. The nanopositioner is bolted to an aluminum fixture with interchangeable bridges used for holding and locating the displacement sensors in the various measurement locations around the straightness reference. Two capacitance sensors, separated by 50.8 mm during EYX , EZX , EBX , and ECX measurements and 19.05 mm during EAX measurements, are

A NEW WAY OF ANALYZING MOTION SYSTEM REPEATABILITY FOR NANOMETER PRECISION: 6-DOF POINT REPEATABILITY

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carriage plane only moves in one dimension because the axis is perfectly straight.

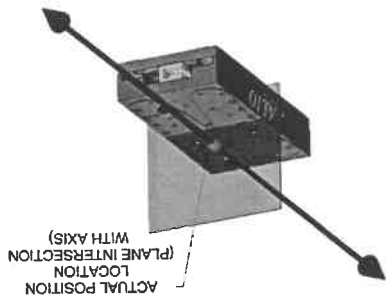


FIGURE 1. The intersection of a plane (fixed to a moving carriage) and an axis is a positional data point used in traditional repeatability testing.

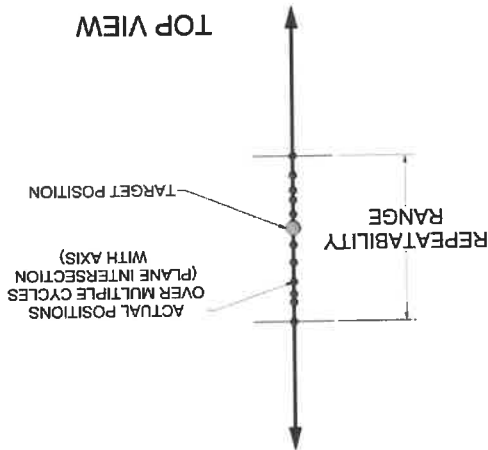


FIGURE 2. Traditional repeatability test method and theory results in a range of points along an axis, which characterizes 1-dimensional repeatability performance.

POINT REPEATABILITY: ACCOUNTING FOR 6DOF ERRORS

In nano-precision systems, often "other" errors that were previously neglected become equal or greater contributors to the six degrees of freedom repeatability performance. Modifying the visualizations above, the axis should actually

Advancements in manufacturing processes and metrology sensors along with the continuing demand to advance technologies and products is driving a greater need for motion systems that are both highly accurate and repeatable at the nanometer level. All motion systems operate in 3-dimensional space and have errors in six degrees of freedom. However, often motion systems are only characterized by performance data of a single or subset of these six degrees of freedom.

ABSTRACT

This practice leaves several error sources unaccounted for in performance data and specifications. Repeatability performance for metrology inspection and manufacturing systems must be analyzed and specified using a point repeatability method that accounts for 3-dimensional spatial errors in order to provide true representation of nanometer precision performance. This paper examines the need for and method of point repeatability testing of single and multi-axis motion systems accounting for six degrees of freedom errors.

PLANE REPEATABILITY: TRADITIONAL SYSTEMS AND TEST METHODS

Many traditional stage and motion systems specify repeatability as a single number representing the variation in linear displacement along an axis of travel. [1,2] Historically, this practice was valid as the repeatability specifications were large enough that other error factors were a small percentage of the total error and could be neglected. This test method can be visualized as testing the axial position (or repeatability) of a plane fixed to the moving carriage, see Figure 1.

In testing, the plane position along the axis is measured over many cycles at a target position. The intersection of this plane with the axis is a point on the axis line and the range of these points results in the 1-dimensional repeatability performance as shown in Figure 2. This test method makes a critical assumption: The

be shown as bending and twisting through 3-dimensional space. As a result, the plane visualization becomes meaningless as the plane will shift, tip, tilt, and twist as the stage moves along the axis. The stage has errors in six degrees of freedom, see Figure 3. Neglecting these additional error sources can result in a misrepresentation of actual stage repeatability performance.

6-D NANO PRECISION™

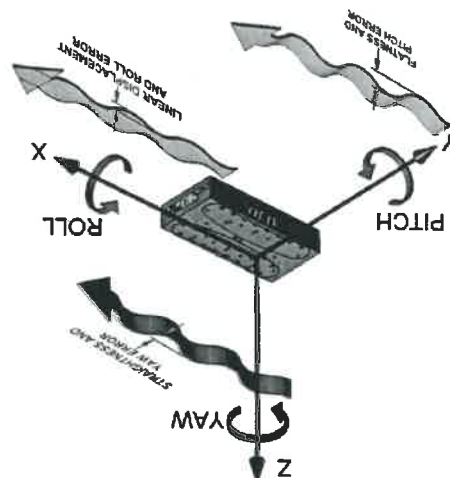


FIGURE 3. The 6 degrees of freedom contributing to repeatability error for a single axis stage.

Each point on a stage mounting surface will move in 3-dimensional space as a result of this error motion in six degrees of freedom. It is the repeatability of this point that must be characterized by testing and specification data. Thus, each point's repeatability will result in a spherical repeatability range in 3-dimensional space, see Figure 4.

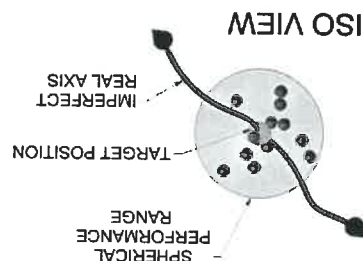


FIGURE 4. A real imperfect axis and the actual spherical point repeatability range.

MEASURING POINT REPEATABILITY

To accurately characterize repeatability, X, Y, and Z components must be measured in a systematic process to characterize the point repeatabilities of a stage along the entire axis.

CONCLUSIONS

With advancements in metrology and manufacturing processes, nanometer precision repeatability of motion systems is in increasing demand. Modern applications not only require repeatability in the axial direction but also require geometric repeatability in 3-dimensional space. In order to have a high confidence in system performance the motion systems must be correctly characterized to account for all error components of stage motion. With proper testing procedures, it is possible to accurately characterize the point repeatability performance at the nanometer level.

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POINT REPEATABILITY TEST DATA

PLANE REPEATABILITY (1D)	POINT REPEATABILITY (6D)
1 DIMENSIONAL LINEAR	1 DIMENSIONAL LINEAR
	FLATNESS
	STRAIGHTNESS
	PITCH
	YAW
	ROLL

TABLE 1. Factors accounted for in traditional 1-dimensional plane repeatability testing compared with proposed six degrees of freedom point repeatability testing method.

Additionally, a process must be implemented to test the influence of pitch, yaw, and roll errors of the axis and their influence on repeatability at critical process point locations. This paper describes a detailed process of measuring point repeatability to characterize single and multi-axis motion systems according to these principles, see Table 1.

A TWO DEGREE-OF-FREEDOM SOI-MEMS TRANSLATION STAGE WITH CLOSED LOOP POSITIONING

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INTRODUCTION

Precise multi-axis positioning is critical to a number of micro- and nano manipulation and probing technologies. Many applications such as fiber optical switches, micro-force sensors, scanning probe microscopy, data storage, micro optical lens scanner depend on such precise positioning capabilities, and for these applications, size, motion range, natural frequency and cross-coupling of motion along the different degrees of freedom are most important characteristics that are directly related to a system's performance. MEMS-scale positioning devices can play an important role in many of these situations. As primary positioning systems, they can reduce the overall system size or, as secondary positioning systems, they can provide a fine positioning/adjustment system very close to the functional element of the system and thereby (i) reduce stacked up of errors, (ii) decrease vulnerability to environmental changes, (iii) improve bandwidth and dynamic performance.

Past research on MEMS-scale positioning systems has focused on arriving at configurations suitable for microscale systems and the fabrication techniques to realize them. Typically, open-loop actuation has been used. This paper addresses two important aspects of MEMS scale positioners: (a) the use of parallel kinematic mechanisms to realize high-bandwidth structures with little or no parasitic motion and (b) the development of integrated closed-loop controls based of capacitive comb sensors.

A Parallel-Kinematic MEMS XY stage

Our research has concentrated on adapting parallel kinematic mechanisms (PKMs), widely used for macro and meso scale positioning systems, to silicon-based MEMS-scale micropositioners. PKMs generally produce high structural stiffness because of their truss-like

structures resulting in higher natural frequencies, and when appropriately designed, can result in configurations where near complete decoupling of the actuation is achieved. PKMs are criticized for the small workspaces or motion ranges because the motion is restricted to the intersection of the motion range of all its kinematics chains. In the context of MEMS and other flexure-based mechanisms, this is not an important limitation, because the range of motion is more severely limited by the elastic range of the flexures themselves.

In the past, our work has introduced electrostatic comb driven, MEMS-scale, parallelogram 4-bar stage designs and used both, linear and rotary combs to actuate them [1, 2], demonstrating de-coupled XY motion of about 20 microns with stage bandwidths of about 1 kHz and q-factors of about 100; three degree-of-freedom stages with two translations and one rotation (XY θ); stages with tilt-plate actuated cantilevers for XY&Z motion [3]; and approaches to actuating and sensing on using a single comb structure [4]. In all our previous work, however, open-loop actuation was used. For applications that did not involve precise positioning under conditions of varying resisting forces, was sufficient. External optical observations of displacement, along with knowledge of the mechanism's stiffness, provided a means of measuring external forces. Such an approach however is limited in both, resolution and speed, and can be applied to static systems in which the signals are large and persistent. In many applications such as tissue and cell testing in mechanobiology and scanned probe imaging where there are changing and unknown disturbing forces, closed-loop control of the axes of the stages becomes necessary.

A schematic of the PKM along with a solid model of its realization of a MEMS device that encodes it is shown in Figure 1. It is

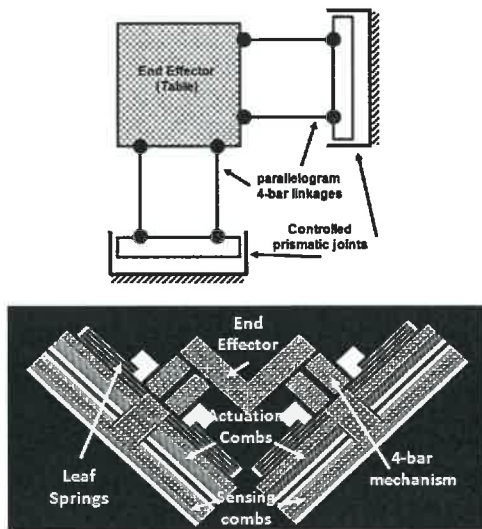


Figure 1: Schematic (top) and MEMS realization (bottom) of a 2-axis translator



Figure 2: First three modes of the system.

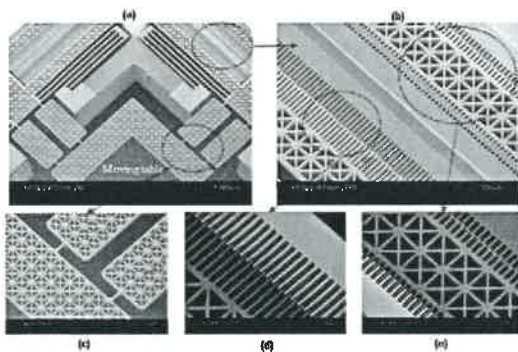


Figure 3: SEM micrographs for micropositioning XY stage. (a) Overall structure; (b) Actuation/Sensing comb drives; (c) Hinges on 4-bar mechanism; (d) Actuation comb drive; (e) Differential comb drives for sensing.

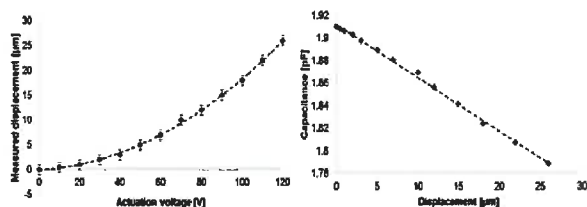


Figure 4: Actuation and sensing characteristics of the fabricated positioning system.

important to note that this PKM has decoupled

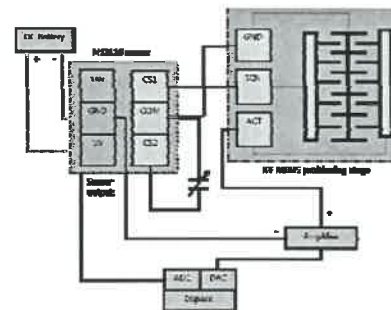


Figure 5: Actuation and sensing circuit schematic

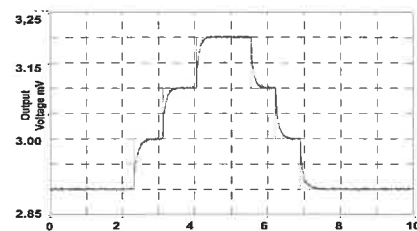


Figure 6: Closed-loop step response of an axis of the positioner.

axes, linear displacement characteristics (at least for the range of motion permissible by the hinge joints, leaf springs and electrostatic comb actuators. As shown in figure 1, the system is actuated by a single electrostatic comb actuator, while the displacement on each axis is sensed by a differential capacitive comb. Also, as shown in Figure 2, the primary modes of the device are translational, suggesting no parasitic rotational motions. The first rotational mode occurs at a frequency around 3 times the two translational modes, suggesting a stiffness of about 10 times for rotational displacements in comparison to translational motions. The device is designed to produce a motion of 20 microns with an actuation voltage of 70V.

The device was realized in an SOI wafer with a 50 micron device layer, 2 micron BOX layer and a 300 micron handle layer using conventional lithographic patterning and dry etching. First the metal contact pads are patterned. Next the mechanism an aluminium pattern is created on the device layer. Following this, the handle layer is patterned and etched using DRIE to provide backside access to the device. The BOX layer is wet-etched next followed by DRIE of the device layer (see [1-3] for details). Figure 3 shows an electron micrograph of the device that is fabricated.

Figure 4 shows the actuation voltage-displacement and displacement-(sensing) capacitance characteristics of an axis of the system. The experimental $\Delta C/\Delta x$ for both axes

was measured to $4.728\text{E-}3 \text{ pF}/\mu\text{m}$. The system is designed to work with a COTS capacitance-to-voltage conversion IC (MS3110 IC from Irvine Sensors, Inc.) that measures capacitances in the range of 0.5-10 pF. Figure 5 shows a schematic of the actuation and sensing circuits.

The system is controlled by PI controller implemented on a D-Space controller. Figure 6 shows the step response as well as the tracking response to a sinusoidal input to one of the axes of the system. Based on the sensing calibrations, in open loop, under ambient conditions, the positioning uncertainty of the system is about 47 nm. With the PI controller, under the same conditions, an uncertainty of about 30 nm was observed.

The system has been designed with dual electrostatic sensing combs to enhance the feedback signal. With the dual combs, the difference in the change of capacitance between the two combs is used. Because one comb experiences a reduction in capacitance, while the other experiences an increase, a differential signal doubles the output voltage. With this configuration, the sensitivity is increased to $0.0136\text{pF}/\mu\text{m}$. With the new sensitivity and some better shielding against noise, the closed loop system bandwidth is about 27Hz and the positioning uncertainty standard deviation is about 5nm. Figure 7 shows the system's response to a step and 1 Hz sinusoidal input.

CONCLUSION:

We have successfully designed and fabricated and tested a closed-loop controlled, 2-axis, MEMS-scale positioning device. This device not only produces nanometer resolution positioning but is also capable of measuring forces down to the nano-newton scale in the x and y directions. Currently, we are working to measure surface tension forces on micron-scale droplets. We are also planning experiments for measuring the adhesion forces between animal cells and different substrates.

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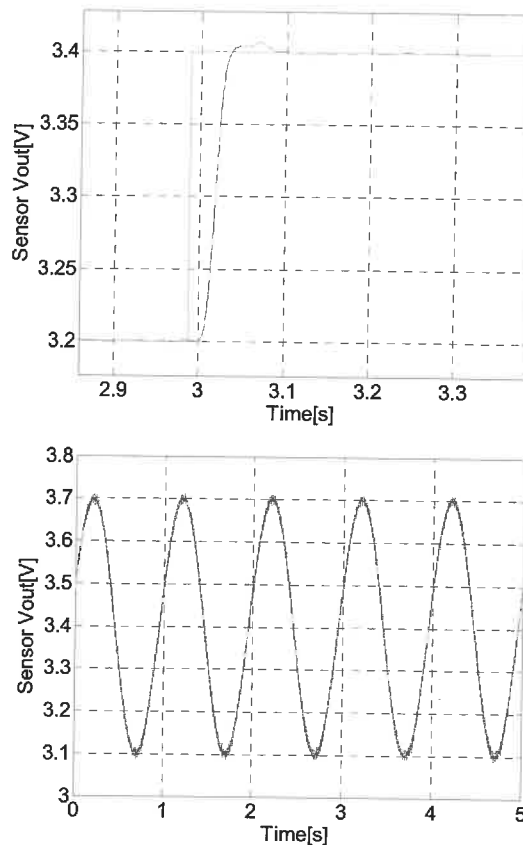


Figure 7: Closed loop response for MEMS XY stage; (top) Step response and (bottom) response to a 1 Hz. sinusoidal reference signal.

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COMPLIANT MICROGRIPPER WITH PARALLEL STRAIGHT-LINE JAW TRAJECTORY FOR NANOSTRUCTURE MANIPULATION

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INTRODUCTION

Compliant gripping mechanisms are particularly suited for precision manipulation of micro- and nanoscale objects by virtue of having zero backlash, no parasitic columbic friction effects, and often only requiring one actuator to close two gripping faces symmetrically about a centerline of action. Many previous compliant gripping mechanisms use a parallelogram structure (Fig. 1a) to close parallel jaws [1,2,3]. These designs are effective in gripping micron-sized objects, but the gripping faces inherently move along an arc-shaped trajectory (Fig.1b). Straight-line jaw motion may be desired for micromechanical tension/compression tests, and for gripping soft objects such as cells, gels, and assemblies of nanostructures such as carbon nanotubes. These and other applications are sensitive to normal and shear loading. In general, the transverse component of the jaw motion constitutes a coupling of two sources of transverse error (δ) in the mechanism: *kinematic error*, defined as the motion path of an equivalent rigid link model, and *elastic error* stemming from finite material elasticity.

We present a compliant gripping mechanism (Fig. 1c) that exhibits a straight-line parallel jaw trajectory. The input (tab) and output (gripper jaw) displacements are proportional, and the proportionality factor can be varied by design. This gripper is a lumped-compliance based flexure mechanism developed from an analogous rigid-link-and-pin-joint mechanism (Fig. 1d). Transverse errors (δ), stemming primarily from linear-elastic structural compliance, are corrected by modifying the mechanism geometry to incorporate kinematic compensation trajectories that redress these errors by superposition. The resulting compliant microgripper features zero *kinematic error*, and the *elastic error to jaw stroke ratio* (δ/d) is minimized to 2.5×10^{-5} . By comparison, a parallelogram configuration of equivalent beam dimensions has a ratio of 4.35×10^{-3} . Further, a parallelogram gripper would require beam

lengths at least one order of magnitude larger than the dimensions of our proposed design to achieve a similar straightness in output path over the same jaw range.

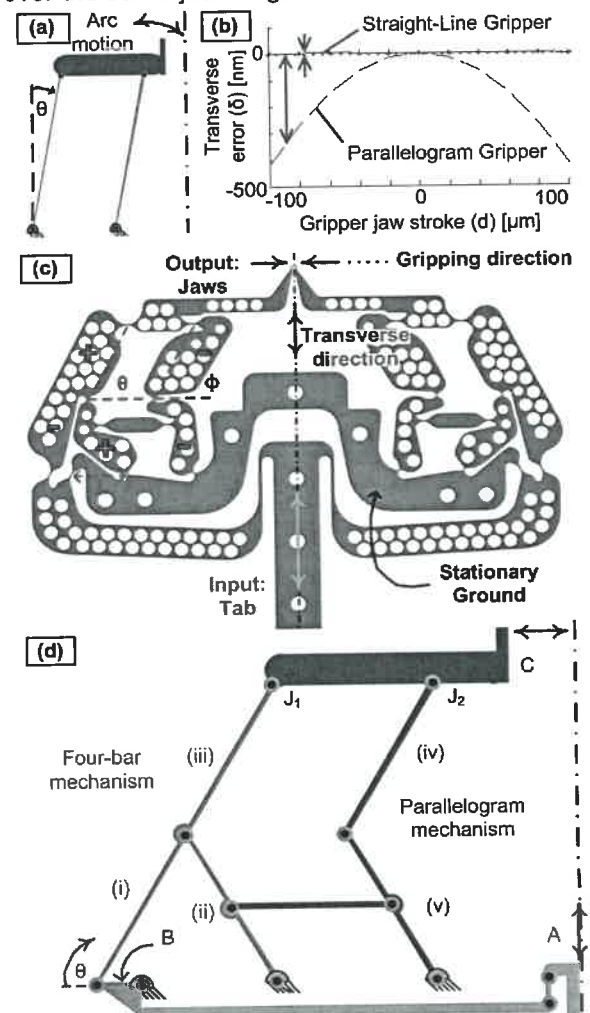


FIGURE 1. (a) Rigid link kinematic model of a typical parallelogram mechanism gripper, (b) comparison of output paths between our straight-line gripper (solid) and a parallelogram (dashed) having equivalent beam dimensions, (c) our straight-line compliant gripper based on an analogous kinematic model shown in (d). (note: mechanisms are symmetric about the centerlines shown).

KINEMATIC DESIGN

The proposed design was conceived as a rigid-link-and-pin-joint mechanism (Fig 1d) comprising 3 parts: a four-bar mechanism that defines the location of J_1 in terms of link B angle (blue); a mechanism that translates the vertical linear motion of the input tab, A, into rotation of link B (green); and a parallelogram-based mechanism that replicates the motion path of J_1 at J_2 , which enables translation of the gripper face, link C, without rotation (red). The four-bar mechanism is a version of the classic "Hoekens" linkage [4] which we optimized for straightness of output path (Fig.2). A notable feature of this linkage is that the angular displacement of link B is approximately linearly related to the translational displacement of J_1 over the stroke range. Further, the maximum displacement of the input tab changes the angle of link B by less than two degrees, which linearly relates input tab and gripper jaw displacements by means of small angle approximation.

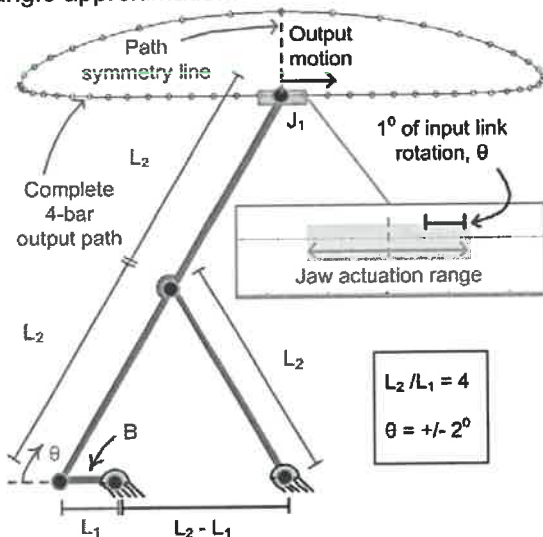


FIGURE 2. Four-bar motion characteristics and beam length relations for optimization

We determined two important relationships between the beam lengths of the four-bar linkage: (1) symmetry of the output path is maintained by relating link lengths L_1 and L_2 as shown in Fig. 2; and (2) the ratio L_2/L_1 determines the curvature of the output path. We wrote a kinematic linkage solver program in MATLAB that iteratively converged upon the ratio $L_2/L_1=4$, for the straightest output trajectory along the bottom portion of the output path. This output path is still an *approximate* straight line in a strict mathematical sense; there is a slight concave-upwards curvature along this bottom

portion. But, this deviation (δ/d) is 5×10^{-11} over the prescribed range of jaw actuation. In terms of the actual gripper dimensions ($L_1 = 5/3$ mm, $L_2 = 6+2/3$ mm), this kinematic error (δ) is 0.01 pm, which is negligible considering fabrication tolerances.

LUMPED-COMPLIANCE MECHANISM

This rigid-link-and-pin-joint mechanism was then translated into a lumped-compliance flexure mechanism (Fig. 1c). Thin compliant hinges occupy points of rotation between significantly wider sections that approximate the rigid links. The hinges have a cycloidal profile [5] which, compared to other compliant hinge contours, maximizes in-plane rotational compliance and translational stiffness for a prescribed angular deflection and material strain limit. Limits of angular deflection ($<2^\circ$) and allowable material strain ($<0.5\%$ for Si) were established for our design.

JAW MOTION OPTIMIZATION

Actuation of the gripper was simulated by two-dimensional non-linear finite element analysis (FEA) using ANSYS. FEA simulations revealed an error ratio (δ/d) of 5×10^{-3} at the gripper jaws. Translational error of the gripper jaw is caused by errors in the trajectory of J_1 . Jaw rotation is attributable to inaccurate replication of J_1 motion at J_2 . These errors may be attributed to two general elastic effects: deflections in the large area geometries, which invalidate the rigid link approximation; and drifts in the rotational centers of the compliant hinges, which invalidate the pin joint approximation.

Elastic error characterization

In our lumped compliance mechanism, stresses develop due to moments exerted by the flexure hinges under bending. To characterize these stresses, we created an analytical model of the four-bar and parallelogram mechanism, using rigid links connected by torsional spring pin joints. We calculated the forces and moments acting at each joint when link B (Fig. 1d) is rotated by a small angle. Using this model, we can understand the development of stresses in the mechanism, which manifest the elastic errors. We determined that, when the gripper is actuated so as to close the jaws, (i.e. downward displacement of the input tab and clockwise rotation of link B), tensile stress develops in the links marked '+' and compressive stress develops in the links marked '-' (Fig. 1c). Since these links and flexure hinges have finite elastic

stiffness, this stress development produces two distinct linear-elastic effects, which we call *parallelogram coupling* and *four-bar loading*.

The tensile stresses in links (ii) and (iii), and compressive stresses in links (iv) and (v) (Fig. 1c,d), constitute a differential loading which, due to material elasticity, produces a 1st order clockwise rotation of the gripping link over this closing jaw stroke as illustrated by the asterisks (*) in Fig. 3a. It is analogous to linear elastic effects in a parallelogram flexure, which have been well characterized and described by closed form equations [6]. Moreover, the tension in links (ii) and (iii) effectuate a 1st order error component in the J₁ trajectory, as shown by the asterisks in Fig. 3b. We call this effect a linear-elastic motion due to *parallelogram coupling*.

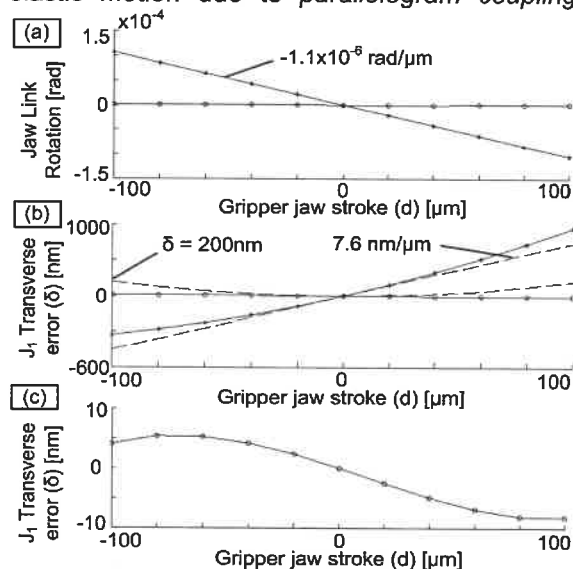


FIGURE 3. (a) Jaw link rotation over stroke for initial (*) and optimized (o) FE simulations; (b) J₁ trajectory over jaw stroke for initial (*) and optimized (o) FE simulations, initial trajectory is also shown decomposed into its 1st and 2nd order components (---).

Compression in link (i) and tension in link (ii) (Fig. 1c,d) generate a 2nd order error component in the trajectory of J₁. We modeled this effect by modifying the MATLAB linkage program to linearly lengthen or shorten the link length during the stroke to simulate compliance. Using this model, we isolated the error contribution for each link, and determined that compliance in links (i) and (ii) can independently create this 2nd order error, and the combination of these contributions is additive. We call this effect a linear-elastic motion due to *four-bar loading*.

Since the compliance in the mechanism is predominantly linear-elastic, we may use the principle of superposition to deconstruct the observed error motion characteristics. Thus, we can isolate the contributions of parallelogram coupling and four-bar loading at J₁ by subtracting a linear fit from the trajectory curve, as illustrated by the dashed lines in Fig. 3b.

Geometric Correction

There are three error motions that must be counteracted to correct the gripper jaw trajectory: 1st order error at J₁, 2nd order error at J₁, and rotation of the gripper link. We modified the linkage geometry to adjust the mechanism's kinematic motion such that linear-elastic errors are cancelled by principle of superposition. Accordingly, these kinematic compensation trajectories are the x-axis reflection of the error motions calculated by the initial FEA simulation (Fig. 3a,b: asterisks), i.e., producing a J₁ 1st order trajectory of $-7.6 \text{ nm}/\mu\text{m}$, J₁ 2nd order trajectory of $-200 \mu\text{m}$ deviation at full stroke, and rotation of $1.1 \times 10^{-6} \text{ rad}/\mu\text{m}$. Consequently, we define and perturb three geometric parameters (Fig. 1c): θ , which changes the 1st order trajectory of J₁; L , which counteracts the 2nd order motion of J₁; and Φ , which counteracts the gripper jaw rotation by introducing a non-parallel angular offset between links (iii) and (iv).

The kinematic model predicts the following modifications to correct the jaw trajectory: $\Delta\theta_{KE} = -0.880^\circ$, $\Delta L_{KE} = -0.7 \text{ mm}$ and $\Delta\Phi_{KE} = -0.386^\circ$. These predictions differ less than 10% from the optimal correction factors determined by iterative geometric modification of the FEA model: $\Delta\theta_{FE} = -0.850^\circ$, $\Delta L_{FE} = -0.7 \text{ mm}$ and $\Delta\Phi_{FE} = -0.354^\circ$. Residual J₁ translational (δ/d) and jaw rotation errors of the optimized mechanism, according to FEA simulation, are 4×10^{-5} and $0.8 \mu\text{rad}$ respectively. Due to the small magnitude and highly non-linear path of this error (Fig. 3c), it may be attributable to elasto-kinematic effects.

FABRICATION AND TESTING

We are working to fabricate a prototype microgripper from a (111) silicon wafer (4" diameter, 500 μm thick) using a through-wafer Deep Reactive Ion Etching (DRIE) process. We chose this crystallographic orientation because silicon has a uniform modulus of elasticity in any direction coplanar with the (111) crystal plane. Therefore, the wafer can be treated as an isotropic material for our case of in-plane bending.

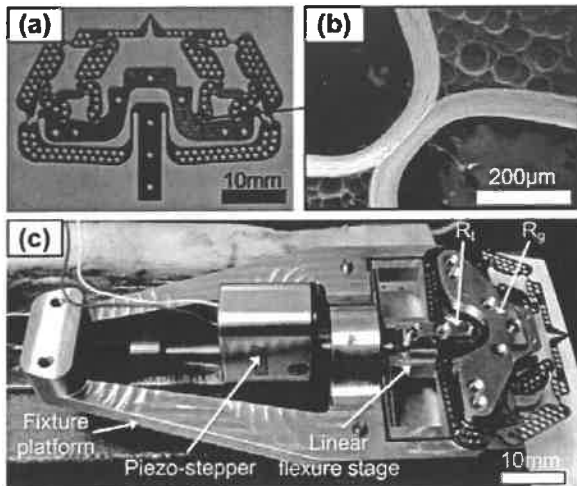


FIGURE 4. (a) Prototype microgripper etched from a (100) silicon wafer, (b) SEM image of joint, showing sidewall due to improper etch recipe, (c) test fixture with piezo actuator and integrated linear flexure stage.

We designed a mask pattern that fits four grippers on one wafer, with each gripper measuring approximately 40 mm by 28 mm. With these dimensions, the jaw range (δ) and error (δ) are predicted to be 400 μ m and 2.5 nm, respectively. For comparison, a parallelogram gripper would require beams greater than one half meter in length to achieve a similar straightness in output path over this jaw range.

The DRIE process generates heat which scales with the size of etch area. Because etch rates are highly sensitive to temperature, thermal variations across the wafer cause non-uniform etching of the gripper pattern. Consequently, we designed the lithography mask pattern such that only a 50 μ m wide kerf line is etched around all of the gripper geometry. We fabricated our first prototypes (Fig. 4a) using an etch recipe that resulted in 2.2 μ m scalloping and 10 μ m/minute etching rate with a selectivity $\approx$ 70:1. However, dimensional accuracy of these microgrippers is poor due to significant tapering of the side walls (Fig. 4b). We are working to optimize our mask patterning technique and etch recipe to achieve nearly vertical side walls with minimal scalloping.

Nonetheless, we have mounted a non-uniformly etched gripper on our custom fixture (Fig. 4c) and actuated the jaws fully closed without breakage. The gripper is secured along the ground link by the top retaining piece (R_g), which is fastened by screws threaded into the fixture platform. The gripper tab is constrained to move

linearly and in-plane by a double-parallelogram flexure stage in the fixture platform. The flexure stage was made by wire EDM. The input tab is fastened to this linear stage by the top retaining piece (R_i). The stage is pre-loaded against, and actuated by, a piezo-stepper with <30 nm step size. We expect this to correspond to <6 nm incremental motion of the gripper jaws. We designed the entire fixture to fit inside a Scanning Electron Microscope (SEM). Once we manufacture a dimensionally accurate compliant gripper from a (111) Si wafer, we will actuate the gripper jaws inside the SEM, and compute the gripper jaw trajectory and incremental step resolution via image analysis.

Because silicon is an extremely brittle material, small defects in the crystal lattice and stress concentrations from rough surfaces can reduce the fracture strain limit from the theoretical value of 4% to <0.2% [7]. Therefore, the actual achievable strain limit is largely based on the quality of the wafer and the etching process. Achieving full closure of the gripper jaws indicates that the DRIE-etched walls should be smooth enough for our target jaw range of 0-400 μ m to be realized in the final prototype.

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INTERFEROMETRIC CONTROLLED PLANAR NANOPositionING SYSTEM WITH 100 MM CIRCULAR TRAVEL RANGE

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INTRODUCTION

The rapid pace of development of the future technologies, like for example nanotechnology, optical high technology or microsystems technology, makes great demands on positioning systems used for inspection and analysis. Objects with dimensions of hundreds of millimeters have to be positioned and measured with precision in the range of nanometers.

Within the collaborative research centre 622 „Nanopositioning and nanomeasuring machines“ at the Ilmenau University Of Technology [1], scientists are working on providing the scientific basis for the necessary nanotechnological equipment. As a part of this project the authors are researching the basics of designing nanopositioning systems for large travel ranges in x and y of hundreds of millimeters. The particular challenge here is to configure the drive system in a way that a position stability within the nanometer range as well as an exact nanometric movement of the positioning stage is realizable despite the large dimensions and masses, which come along with the large operating areas. At this point, conventional systems with roller guides and stacked linear axes reach their limits. Long kinematic chains, mechanical resonances and last but not least, frictional forces within the guiding elements rank among the critical parameters and limit the reachable precision.

PLANAR DIRECT DRIVE PRINCIPLE

A planar aerostatically guided direct drive system offers completely new options compared to conventional systems and offers the best preconditions for a nanometric precise movement with minimal friction [2], [3]. The configuration of an integrated direct drive system is shown schematically in *FIGURE 1*. The slider is supported almost frictionless by planar air

bearings and can move in x , y and φ_z . The integrated direct drive is formed by three linear drive units, each consisting of a pair of frame-fixed flat coils and a permanent magnet array mounted to the sliders underside. The three drive units each generate a drive force perpendicular to the direction of the coil.

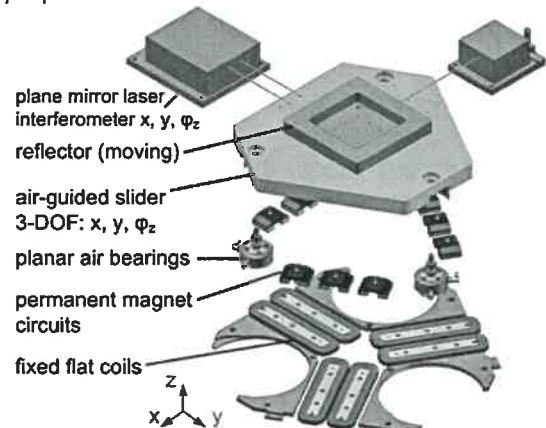


FIGURE 1. Principle of the planar drive system.

During regulated operation the three forces act simultaneously on the slider and due to the 120° arrangement of the drive units the resulting force can be pointed to any direction in the xy -plane. In addition a torque around the z -axis can be applied to control the slider rotation. Hence the movement of the slider can be controlled in x , y and φ_z while it is measured with a high-resolution single beam (x) and a double beam (y , φ_z) plane mirror interferometer [4]. In this form the simple kinematic structure, with the direct driven slider as the only moved part, enables very good positioning capabilities.

DESIGN OF THE PLANAR DIRECT DRIVE

Based on this principle a planar positioning system was developed where the slider is made of zerodur and has zerodur reflectors bonded to it. Thereby the slider of the direct drive system

and the reflector for the laser interferometers are incorporated in only one rigid moving body, thus enabling a high stiffness in the actuation chain. Furthermore this reduces the problem of connecting different materials and provides a thermally stable mirror mount for the position measurement. In order to achieve a high stiffness in z-direction vacuum-preloaded air bearings are applied to support the slider on the granite base. *FIGURE 2* shows a photograph of the realized system.

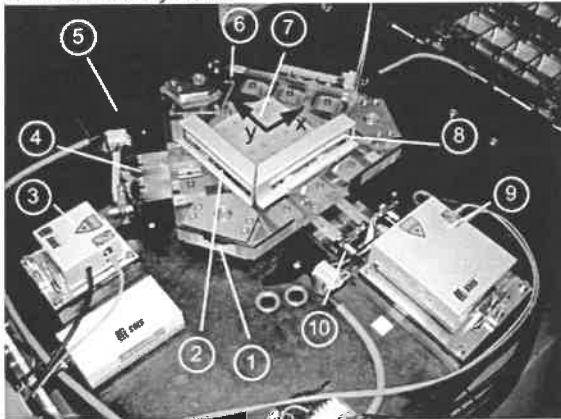


FIGURE 2. Planar Nanopositioning System.

- | | |
|--------------------|-------------------------------|
| 1 air bearing | 6 zero-dur slider |
| 2 x-reflector | 7 object table |
| 3 x-interferometer | 8 y, ϕ_z -reflector |
| 4 drive coils | 9 y, ϕ_z -interferometer |
| 5 granite base | 10 capacitive probes. |

The control loop is closed by a rapid control prototyping system dSpace DS1006 where the control algorithm is implemented as a real time application running at 10 kHz sample frequency. In the current configuration the realized system has the following characteristics:

- | | |
|------------------------------------|----------------------------|
| • Travel range in x, y | $\varnothing 100$ mm |
| • Resolution of measuring x, y | 0.1 nm |
| • Acceleration in x, y | max. 500 mm/s ² |
| • Positioning velocity | max. 30 mm/s |
| • Resolution of measuring ϕ_z | 0.001 arcsec |
| • Moved mass | 9.6 kg |

INITIALIZATION OF THE LASER INTERFEROMETERS WITH A FLOATING SLIDER

The slider of the planar positioning system has no mechanical fixture to prevent slider rotation and besides the air tubes no mechanical contact to the stator. This is positive for precise positioning but it is a challenge when working with laser interferometers. That is because for the interferometer initialization the reflected beam must be coincident with the outgoing beam, which means that the slider with the

reflectors must be precisely aligned to the stator fixed measuring heads. The angular tolerance is ± 30 arcsec only. To achieve this, a constant current is applied to the three drive units so that the resulting forces lead to a centering of the slider with respect to the drive coils. Experiments show that with a centering current of $I_c = 1.5$ A the oscillation of the slider position is less than ± 1 μ m in x- and y- direction and less than ± 1 arcsec in ϕ_z (see *FIGURE 3*). The experiments proved that this centering position can be precisely reproduced so that the interferometers can be initialized and the controller can be taken into operation.

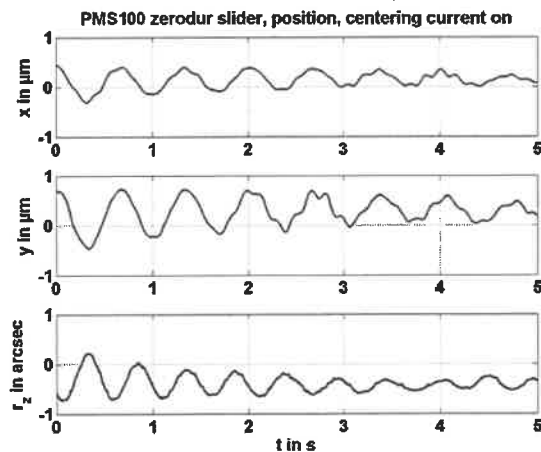


FIGURE 3. Position oscillation with centering current $I_c = 1.5$ A

POSITIONING IN THE NANOMETER RANGE

With the position- and rotation controller in operation, movements in the nanometer range can be performed. Due to the advantageous mechanical characteristics and the stiff actuation chain a stiff controller can be applied resulting in a high bandwidth and thus very good positioning capabilities. Very small internal disturbances, especially the lack of friction forces also contribute to this.

In this configuration the motion system serves as a test bench for a detailed analysis of the positioning capabilities. Therefore another interferometer was added to measure the z-position of the slider simultaneously. The interferometer head is mounted to a portal bridge while the reflector is positioned on the object table of the slider. Thus the z-vibration of the slider caused by the aerostatic slider support or other disturbances can be measured with a high resolution. *FIGURE 4* shows the principle and the measuring circuit of the z-measurement in a sectional view of the CAD-design.

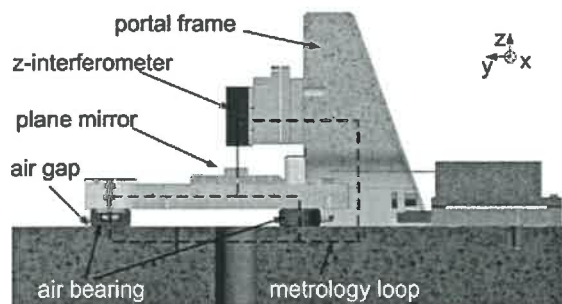


FIGURE 4 Measurement of the z-position

In order to characterize the set-up at first the position of the slider was measured with air bearings turned off and the controller not in operation. This was done to quantify the present disturbances and the measurement noise. For the measurement the position signals from the x-, y- and z-interferometers were traced for $t=5$ s with 10 kHz sample frequency. The standard deviation for these signals was calculated to $\sigma_x=0.41$ nm, $\sigma_y=0.33$ nm and $\sigma_z=0.36$ nm. With the system taken into operation and commanded to target position $(x,y)=(0,0)$ the experiment was repeated and the measured data is shown in FIGURE 5. As a measure of the drive systems capability to achieve the commanded position, the servo error was calculated as a root mean square (RMS) value of the position error. The servo error (RMS) in this case is $e_x=0.33$ nm and $e_y=0.45$ nm respectively. The third subplot presents the simultaneously measured z-vibration of the slider, which proved to be also in the range of nanometers: the standard deviation calculated from the z-time series is $\sigma_z=0.45$ nm.

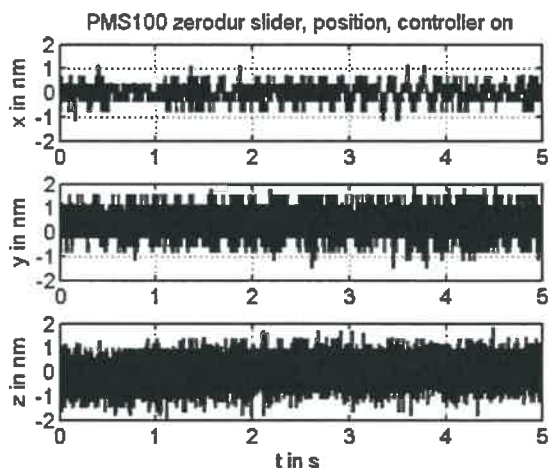


FIGURE 5. Standstill performance with slider lifted and controller in operation

Comparing the achieved servo error in standstill operation with the noise values measured with a deactivated slider we find that with this performance we are at the limits of what can be reached with this set-up. The present servo error is mainly caused by the present noise in the set-up and a further reduction of the servo error will only be possible if this noise can be reduced considerably e.g. by reducing measurement noise.

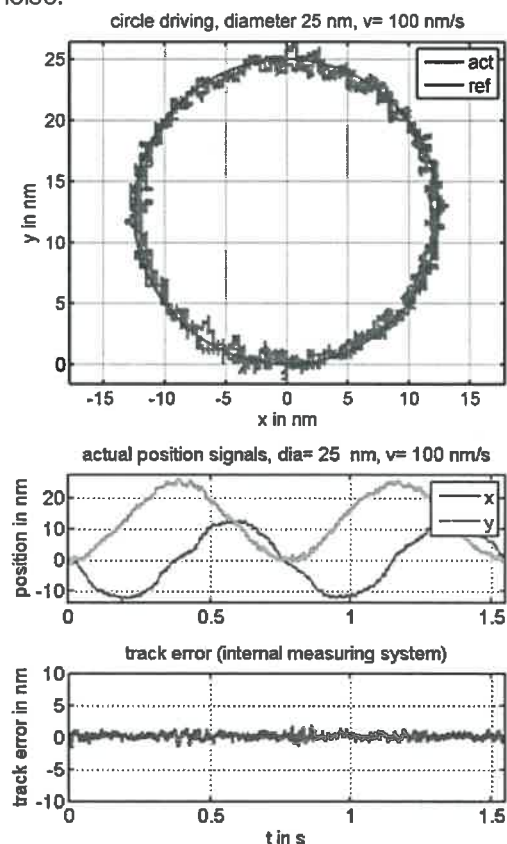


FIGURE 6. Measuring data of a circular driving (dia = 25 nm, $v = 100$ nm/s).

Besides the characteristics of the positioning system in standstill operation it was also investigated how good the slider can be forced to follow a target path. Therefore experiments were carried out with a circular target path with varying diameters and positioning velocities. FIGURE 6 shows the result of tracking a circle with a diameter of 25 nm with a velocity of 100 nm/s. As a performance measure the track error is calculated as the deviation of the actual slider position (measured with the internal x- and y-interferometers) from the required position on the target path. In this experiment a maximum tracking error of $e_{\max}=2.0$ nm was observed.

These performance measurements illustrate the positioning capabilities of the integrated direct drive system. Particularly they prove the advantages of the almost frictionless aerostatic slider support as there is a smooth passing through the reversal points of the axis movements, where roller guided systems usually experience problems due to the alternating friction forces.

ANALYSIS OF THE SYSTEM DYNAMICS

For a detailed analysis of the system dynamics the frequency-response-function (FRF) was measured in the x-, y- and z-direction. The intention is to compare the measurements with the results of a system model and thereby identify the critical parameters that have strongest influence on the dynamic behavior and thus determine the system performance. In a first step a 3-DOF model with a single mass and two springs representing the support stiffness was built up to simulate the system in x, z, ϕ_y and y, z, ϕ_x respectively (see FIGURE 7).

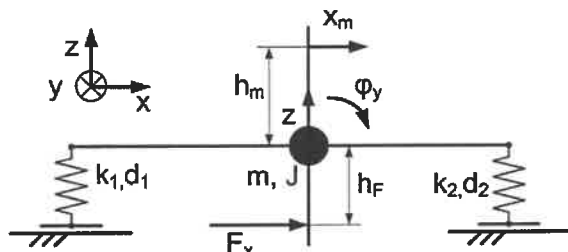


FIGURE 7. 3-DOF model of the mechanical system for simulation of the FRF in x-direction

Looking at the x-direction exemplarily, the system input is the drive force F_x while the measured x-position x_m represents the output of the system. The measured FRF for the x-direction is shown in FIGURE 8 together with the FRF derived from the model. It can be seen that the transfer behavior is dominated by the -40 dB slope of the double integrator and that there are only little interior dynamic effects. This justifies the approach with the single mass model as a first step. With this simple model a part of the measured system dynamics can be identified: the resonance at about 320 Hz resulting from the tilting of the slider on the supporting springs. Via this resonance frequency the air bearing stiffness appears in the system transfer behavior. With the help of the model it is now possible to estimate the change in the transfer behavior for a varying air bearing stiffness. However the model has to be upgraded in order to cover a larger part of the system dynamics.

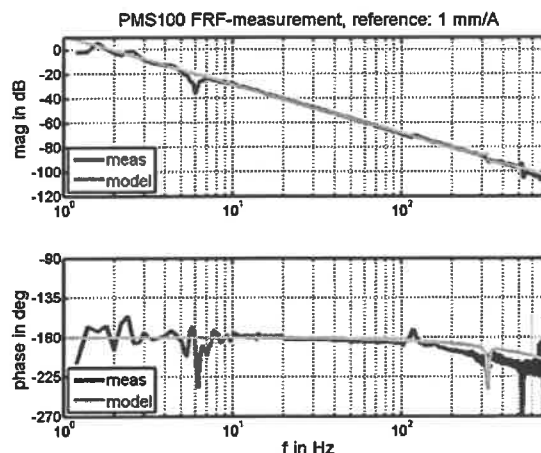


FIGURE 8. Bode plot of the measured and the modeled FRF in x-direction

Future Work

The detailed investigation of the system dynamics must be continued in order to develop a sophisticated model for the estimation of the transfer behavior based on given design parameters like for example the air bearing stiffness or the stiffness of the slider. This will be particularly useful when designing new systems because it allows an estimation of the transfer behavior already in early design phases.

Acknowledgements

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NON-LITHOGRAPHICALLY-BASED MICROFABRICATION OF PRECISION MEMS NANOPositionING SYSTEMS

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INTRODUCTION

Microelectromechanical Systems (MEMS) positioners offer many potential benefits over macroscale systems due to their high speed, low cost, integrated sensing and fine resolution. However, MEMS production has not had the same success as photolithographic integrated circuit (IC) production due to its high capital costs coupled with its generally lower market demand. The purpose of this work is to develop a non-lithographically-based (NLB) microfabrication process chain that can rapidly (≈ 1 day) produce low cost ($\approx \$100/\text{device}$ fabrication cost) MEMS positioners with high resolution sensing. This new process makes it practical to fabricate MEMS positioners, such as the Hexflex-based design envisioned in Figure 1, in small production volumes without the prohibitively large cost and time investments associated with present MEMS fabrication. This will allow rapid research and development cycles for new MEMS positioning equipment, which could enable advances in a range of fields including nanomanufacturing, personalized medication, computing and data storage, and metamaterials-based energy generation/storage.

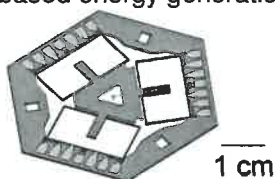


FIGURE 1. Proposed hexflex-based nanopositioner structure with integrated sensing.

BACKGROUND

MEMS positioners generally utilize integrated sensing, which requires $\mu\text{m}/\text{nm}$ -scale electrical structures to be created on a nanopositioner [1]. In order to do this, it is necessary to be able to i) bulk micromachine cm-scale mechanical structures, ii) surface micromachine integrated $\mu\text{m}/\text{nm}$ -scale electrical features, and iii) do this all at low average cost ($< \$1\text{k}$) per device for the low production volumes typical of research and development. Unfortunately, no established

fabrication process chain is able to simultaneously meet all three of these requirements. Traditional macroscale machining can produce cm-scale parts from robust materials at low cost, but cannot produce $\mu\text{m}/\text{nm}$ -scale electrical features. IC photolithographic fabrication has been used to produce meso-/micro-scale nanopositioners, [1] however it results in high ($\$10\text{-}100\text{k}$) average device costs for small batches and brittle structures that are unsuitable for field use.

A range of non-lithographic (NL) fabrication processes have the potential to meet all of the conditions above [2,3]. Previous work has been carried out on bulk micromachining of metal mesoscale mechanical structures [2] as well as electrical structures [3]. However, these efforts have not successfully been integrated on a single device due to fabrication difficulties at the interface of the two types of structures. In this paper we will demonstrate a NLB process chain that is capable of meeting bulk/surface micromachining and cost requirements.

PROCESS OVERVIEW

The NLB microfabrication process described in this paper can be broken down into four main steps: i) bulk micromachining ii) surface micromachining, iii) thin film patterning and transfer (TFPT), and iv) circuit bonding.

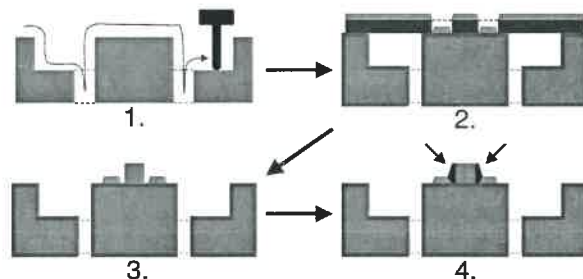


FIGURE 2. NLB microfabrication flow: (1) bulk micromachining, (2) surface micromachining, (3) sensor placement and (4) circuit bonding.

The mechanical structure is formed during bulk micromachining using a micromill. The electrical structure is formed during surface micromachining using type II sulfuric anodization and e-beam physical vapor deposition through a mechanical shadow mask. The sensors used in the device are patterned and attached during TFPT, using a laser scribe to shape silicon piezoresistors and a metal stamp to place them. The sensors are attached to the electrical traces during the circuit bonding step, using specialized metal bonding to the silicon in conjunction with a conductive epoxy.

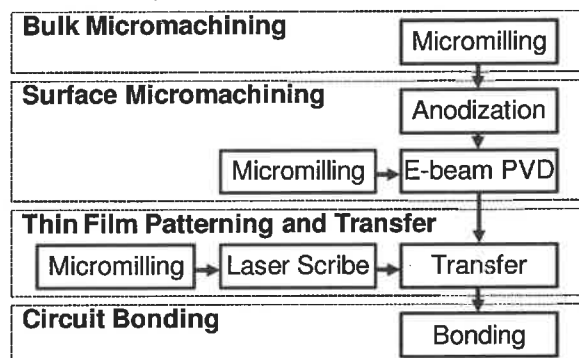


FIGURE 3. Detailed process layout.

A sample device was fabricated to demonstrate the feasibility of the new process chain. The device consists of a simple cantilever with a 50 μ m thick silicon piezoresistor bonded to the flexure. 500 μ m wide, 1 μ m thick aluminum traces are used to connect the ends of the transducer to bond pads.

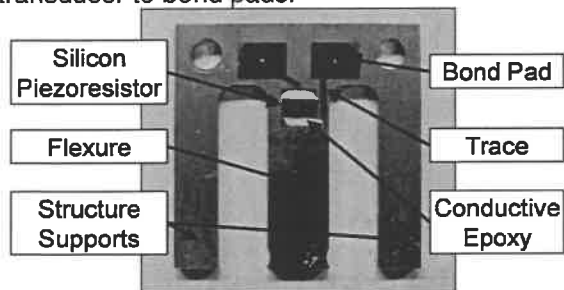


FIGURE 4. Mesoscale electromechanical system composed of a simple cantilever and a silicon piezoresistor, fabricated using NLB methods.

Order of Operations

The order of the steps and the processes used are determined by the fabrication requirements such as throughput time, rate, fixed cost, variable cost, and batch size. The process development described in this paper focuses on the extreme end of the range of requirements, as determined by research and development.

These conditions are: short cycle time and low fixed cost, while variable cost may be high and rate may be low. The specific processes used for each step reflect these requirements, as they are chosen for their high flexibility and low fixed cost. Higher volume operations could be carried out with processes like chemical etching, which bring a higher fixed but lower variable cost.

Bulk Micromachining

The overall mechanical structure of the device is generated in the bulk micromachining step. This is carried out first in order to avoid issues with protecting the fragile surface electrical structures. This also allows for the use of more aggressive processes like chemical etching.

The bulk micromachining operation must be capable of putting small ($\approx 10\mu$ m) features in cm-scale structures made of a range of materials. A Microlution 3-axis micromill was chosen due to the flexibility of micromilling with regards to both structure and material. The micromill is also capable of fine feature generation. The aluminum stock is attached to the micromill pallet with Crystalbond 509 removable wax adhesive. This rigidifies the part during the cutting, even when shaping small structures.

Surface Micromachining

The electrical sensor structures are generated in the surface micromachining step. This is broken into two operations. The first step is to isolate the conductive substrate (for metal structures) from the soon-to-be-deposited electrical surface features. The second step is to create these electrical structures.

The isolating operation must create a mechanically anchored, insulating surface for the electronics. Type II sulfuric acid anodization of an aluminum structure was chosen due to the ease of producing a dielectric layer over a machined surface that would adhere strongly to the surface and could be made of sufficient thickness (5-10 μ m) to isolate the electronics. This process can additionally be done with low time and cost investment. Anodizing is specialized to aluminum, but the isolation operation may be adapted to non-aluminum structures by depositing aluminum over the structure through ALD, PVD or electroplating [4], then anodizing this layer. The mechanical structure may alternately be made of an inherently non-conductive material (e.g. plastic), removing the need for this step entirely.

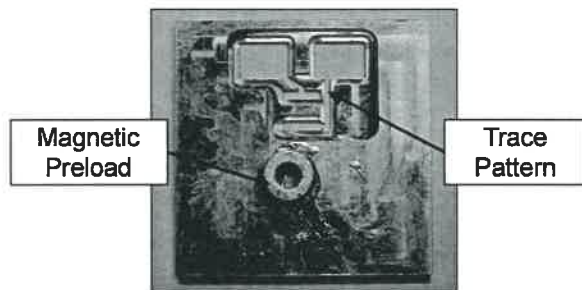


FIGURE 5. Mechanical shadowmask used to form conductive traces on the test structure shown in Figure 4.

The trace deposition step must be capable of creating traces down to about $\approx 50\mu\text{m}$ width and $\approx 1\mu\text{m}$ thickness over machined surfaces. These values derive from the need to put multiple traces along a flexure and to avoid altering the device mechanical properties with the traces. E-beam PVD through a mechanical shadowmask was chosen due to the ability to put down thin films of high quality [5]. A mechanical shadowmask is because it offers several advantages. First, metal shadowmasks can be fabricated using micromilling down to the required feature size for significantly lower cost and time than a standard photolithography mask. Second, the shadowmask can also be made conformal to the arbitrary 3D geometry of the positioner, should extra features be machined into the face of the positioner. Third, the patterning of the surface is done mechanically rather than chemically, which means the complex process research to determine process and material compatibility, typical of MEMS development, can be bypassed. This process is scalable to higher volumes. An alternative process for higher flexibility is aerosolized spray printing [6].

Thin Film Patterning and Transfer

The sensor transducer is fabricated and attached to the mechanical structure in the TFPT step. This step is driven by the use of bulk silicon piezoresistors. Other strain gauge materials like polymers [7], metals or CNTs [8] might be substituted for silicon. Bulk silicon piezoresistors are used in this research due to their high gauge factor and low flicker noise at the mm-scale. This process is broken into two steps: patterning the bulk silicon while it is attached to a stamp, and transferring the patterned structure to the device.

The patterning operation must be able to create $\approx 50\mu\text{m}$ features out of a thin silicon wafer with

the correct location and orientation on a stamp. The thin wafer is required to avoid having the silicon dominate the mechanical properties of the flexure on which it is located. The alignment and location of the structures on the stamp avoids the need for directly handling the small silicon piezoresistors. A wafer laser scribe (Electrox Scorpion Rapide, 20W Yb:Fibre laser, 1063nm) was chosen to cut $50\mu\text{m}$ silicon wafers which are adhered to a stamp using Crystalbond removable adhesive. This laser scribing operation can meet all requirements as it has a kerf width of approximately $20\mu\text{m}$, and due to the serial nature of the process, it can be altered for new designs with minimal time or cost investment. The stamp is a glass slide, which may be attached to an aluminum backing to give a coordinate frame for accurate cutting. The glass slide is not affected by the wafer cutting laser. Aluminum was found to pit during cutting and weld the silicon to the stamp.

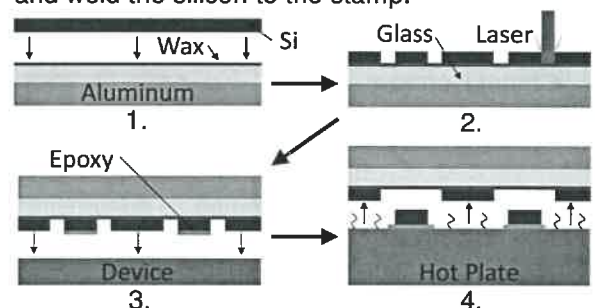


FIGURE 6. TFPT process flow: (1) Silicon adhesion, (2) laser patterning, (3) aligned stamping and (4) stamp release.

The transfer operation must be able to imprint the patterned silicon onto the device with a placement accuracy of $\approx 10\mu\text{m}$. The operation of pseudo-kinematically aligned stamping in conjunction with a thermoset adhesive (e.g. Epoxylite E8121) is used because the alignment and bonding operation can be parallelized for all piezoresistors. This reduces the complexity of the operation for multi-sensor systems.

Circuit Bonding

The sensor transducer is attached to the previously generated electrical circuit in the circuit bonding step. This step focuses on the interface between the sensor material- silicon- and the trace material- metal.

The bonding operation must electrically link the piezoresistor to the traces with a contact resistance about 10% of the piezoresistor. This

process is still under study due to the difficulty in engineering metal-semiconductor interfaces to minimize the Schottky barrier effect. Early results suggest that properly designed contacts using rare earth solder (Adhera Technologies AS-3EC) or Indium solder can produce a near-ohmic contact with resistivity around 5-20% of the silicon resistance. The soldering method offers the potential for a simple, rapid means of making the necessary contact. This is compared to the more operationally complex method of deposited metal bond pads, which require wafer cleaning, patterning and annealing steps.

RESULTS

Two main limitations were observed with the testing structure: the metal-semiconductor interface performance and the deposition of metal traces on a roughened surface.

The metal-semiconductor interface must be properly engineered to avoid overwhelming the piezoresistor with a large contact resistance, which reduces the effective gauge factor. Rare Earth and Indium solders appear to be capable of breaking through the surface oxide and diffusing into the silicon [9], thus reducing this contact resistance.

The patterning of continuous conductive and insulating films onto the mechanical structure proved to be dependent on the surface quality. Surface micromachining of thin films produces poor quality films on roughened surfaces [5] because roughness based variations in the surface potential drive an uneven distribution of atoms. These roughened surfaces are found on the bulk-micromachined materials used in this fabrication process. A set of rules that describe how to adjust designs and fabrication processes to ensure quality film deposition are required to fabricate reliable devices.

CONCLUSION

The new fabrication method presented in this paper is capable of rapidly fabricating robust low cost MEMS with micron scale features and cm scale sizes. This process utilizes a range of NL fabrication techniques to carry out: (i) bulk micromachining, (ii) surface micromachining, (iii) TFPT and (iv) circuit bonding. The focus on NL fabrication processes enables a wide range of materials to be used, increasing the design freedom for MEMS devices. High resolution silicon strain sensors are used in this process.

These sensors have a gauge factor around 60x that of metal foil gauges, enabling dynamic ranges above 100dB. Advances in materials and fabrication techniques will also increase the number of sensor and actuator types that can be used in NLB fabrication. The two major limitations of the process lie at the interfaces of the traces with the roughened surface of the structure and with the silicon piezoresistor. Further study of the process and design parameters affecting these limitations should improve the reliability of the process chain.

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FLEXURE-BASED, PARALLEL NANO-POSITIONING ROBOT

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OVERVIEW OF THE HEXAPOD DESIGN

This paper describes the design of a novel, parallel robot with flexural joints configured as a hexapod. The mechanism is intended as an automated, 6 degree of freedom, nano-position adjustor for opto-mechanical and other high precision adjustment applications. Several interesting features, including a flexural locking mechanism and fixed length legs with flexural bearings, are included in the design. The flexural locking mechanism provides excellent stability after the position adjustments have been made. Unlike most commercially available hexapods which employ rotary bearings, the hexapod developed here has fixed length legs with flexural joints on either end. The monolithic construction of the flexures eliminates friction, backlash, and other mechanical losses within the hexapod maximizing resolution and simplifying assembly. The base of each fixed length leg is guided by a linear bearing. The linear bearing, in turn, is coupled to a high resolution linear actuator and high resolution absolute position sensor. The linear actuators are non-rotating traction driven screws actuated by nonresonant piezo rotary motors; the actuators are low cost, off the shelf items. These actuators have 30 nanometer minimum incremental motion capability and power-off position holding. The absolute position sensors used are six LVDTs.

The fixed length leg arrangement allows the mechanism's actuators and position sensors to be disconnected and removed following the completion of a position adjustment and locking. Together, the actuators and sensors form a piece of tooling that can be used to adjust many mechanisms in a production setting (thereby controlling costs). The combination of flexural bearings, fixed length legs, flexural locking mechanism, and removable actuators/sensors results in a novel, high performance, low-cost positioning mechanism.

A control program developed in Visual C++ contains forward and inverse kinematics for the hexapod mechanism and allows adjustments to be made in Cartesian (X, Y, Z, RX, RY, RZ) coordinates of the output stage. Testing of a prototype mechanism has demonstrated output positioning resolution of 80 nm and 3 μ rad (which is limited by the resolution of the LVDT position sensors). Furthermore, dynamic measurements of the mechanism, when in a locked position, show a first eigenfrequency of over 560 Hz. Compared to more traditional serial adjustors, the hexapod mechanism is simpler, easier to adjust, less expensive to fabricate, and easier to lock in position following adjustment.

Figure 1 shows an embodiment of the proposed hexapod-based adjustable mount for an optical component. Each leg has flexures arranged to provide compliance in all degrees of freedom except axially (each leg is compliant in the other 5 degrees of freedom). The six legs fully define the platform position without over-constraint.

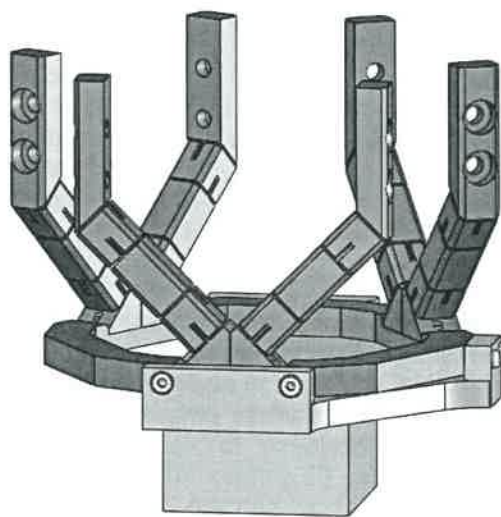


FIGURE 1. Fixed length leg hexapod with flexural bearings arranged such that each resulting leg is axially stiff.

The hexapod is adjustable when the base of each leg is allowed to translate in its single degree of freedom. Figure 2 shows the 6 flexures mounted on linear bearings allowing the base of each flexural leg to be translated vertically. The linear bearings shown in the figure are rolling element bearings (crossed roller), but linear flexure bearings are also possible. The figure also shows 6 axially stiff wire-type flexures mounted on each linear bearing. These flexures allow the connection of both a locking mechanism and a removable actuator for adjusting position. The off-axis compliance of these flexures helps minimize the influence from the lock or actuator in any direction other than axial.

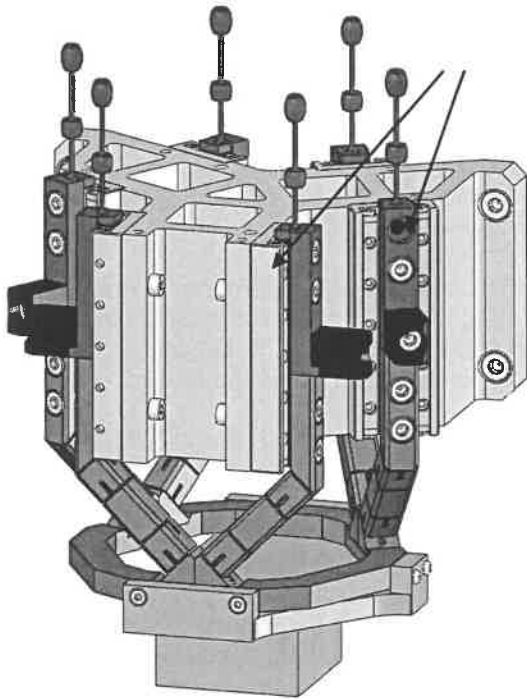


FIGURE 2. Hexapod mounted on 6 rolling element linear bearings. Axially stiff flexures (in blue) allow the non-influencing connection of actuators (not shown).

Figure 3 shows the linear actuators and locking clamps interfaced to the wire flexures. These are nonresonant piezo-type actuators (Newport PZA12) that employ a traction driven lead screw arrangement with rotating nut and translating screw (the output) to give a 30 nm incremental motion capability. Additionally, LVDTs (linear variable differential transformer) are provide absolute position feedback on the actuator displacements.

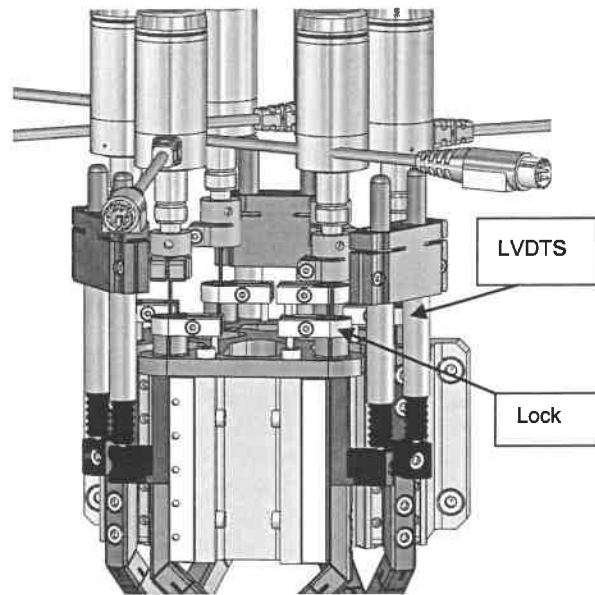


FIGURE 3. LVDTs provide absolute position feedback on position of each flexure leg.

The locking mechanism fixes the hexapod's position after adjustment is complete. An additional coupling attaches the actuator to the axially stiff wire flexure connected above the linear bearing. This allows the actuators to be decoupled and removed from the assembly as a single unit. In this manner, the actuators can be used as removable and interchangeable tooling and do not need to be fabricated for every production unit. The locking mechanism is a flexural collet that when pinched closed provides sufficient pinching force. The pinching force is lateral or orthogonal to the stiff axis of the locking flexures. This means that any lateral disturbances are attenuated by the lateral compliance of the flexure and do not directly influence the resulting position of the mechanism.

The primary benefits of the design may be summarized as:

1. Six flexural legs are fixed in length to simplify construction (commercially available hexapods or Stewart platforms and those described in the literature have variable length legs).
2. The design uses flexural bearings as opposed to sliding or rolling contact rotary bearings such as u-joints or spherical joints. This design approach eliminates friction within the hexapod maximizing resolution and simplifying construction.

3. The design can use a variety of linear actuator types including piezoelectric, voice coil, or linear motors.
4. Actuators are fixed to ground and are not packaged within the hexapod legs. This key innovation allows the hexapod frame to be made very compact.
5. The design includes a mechanical locking technique that creates high stability in the hexapod position and allows power to be removed from the actuators.
6. In the design, the actuators are separate tooling that may be removed after position adjustments are made. After the locks are engaged and actuators are removed, the flexure based hexapod becomes a dynamically stiff, static structure supporting the payload at the desired position.
7. The design includes a locking mechanism that minimizes influence of the position adjustment when the lock is engaged. The lock also maximizes stability of the resulting hexapod position.
8. Compared to many serially arranged adjustment mechanisms, the parallel hexapod offers improved dynamics (higher resonant frequencies). Error propagation is minimized compared to serial adjustment mechanisms.

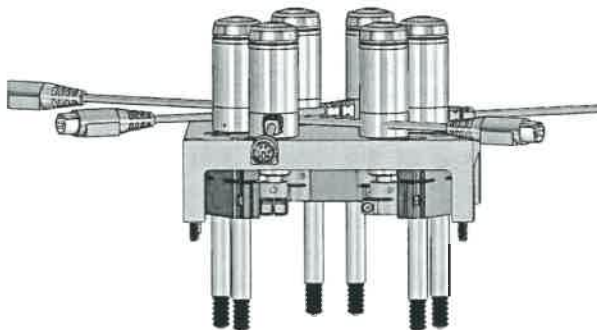


FIGURE 4. Removable tooling plate with actuators and LVDT position sensors.

To fulfil the travel range requirements the flexures within the hexapod must deflect sufficiently to accommodate the needed actuator displacements. To analyze the travel range, a full kinematic model of the hexapod created in MATLAB was used to calculate the actuator displacements required to reach the extremes of travel in all degrees of freedom simultaneously.

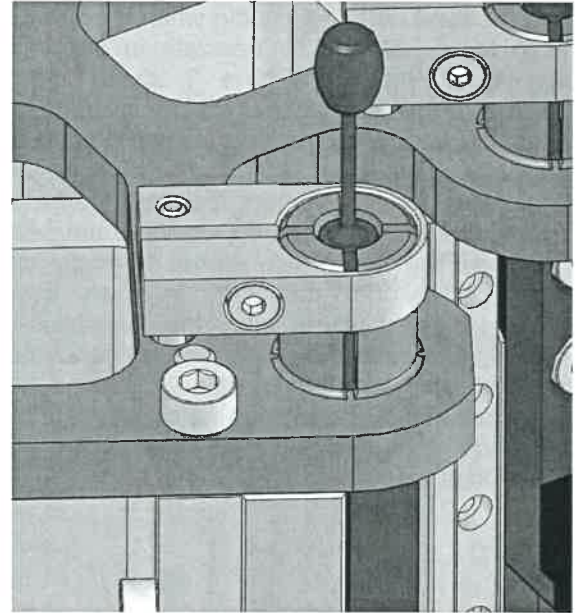


FIGURE 5. Detail view of position locking mechanism. The split collet closes to lock the position without axial influence.

These actuator displacements were, in turn, used as displacement constraints within a static FEA model of the hexapod to calculate maximum stresses within the flexural elements.

For the hexapod, the worst case occurs for an RX, RY rotation point located at 108 mm below the prism. The large goniometric moment arm results in relatively large motions of the hexapod. At the extremes of travel, the actuators must displace up to ± 2.3 mm. Under these conditions, maximum stresses in the flexures which are fabricated from solution heat treated titanium alloy, Ti 6Al4V, reach approximately 80% of yield stress. Because the adjustment process is low cycle operation, there is minimal exposure to fatigue failures. For a rotation point centered within the prism, travel ranges for the actuators are reduced to ± 0.6 mm and the maximum stresses are less than 20% of yield.

The motion requirements of the hexapod and the rotation point located 108 mm below the prism result in a workspace that is basically a cylinder that is 1.85 mm radius by ~ 0.7 mm tall (that is ± 0.35 mm relative to the nominal center position). Most of the Z workspace height is driven by the RX/RX motion and comes at the outer edges of a workpiece surface. For visualization purposes, imagine that this cylinder is swept out by a point at the center of the

workpiece as the hexapod moves throughout the workspace.

In addition to the stress analysis, the static FEA model was used to confirm the kinematics of the flexure hexapod. This effort begins with the MATLAB model of the hexapod to calculate actuator displacements required for a desired motion in X, Y, Z, RX, RY, and RZ. A full FEA model of the flexure hexapod was used to confirm the kinematics. Each leg is constrained so that Z displacements can be input at the actuator feet while the other five degrees of freedom are constrained to zero. The static FEA then shows the resulting displacement of the hexapod matches the desired position within a few microns even for very large displacements (very low coupling).

In addition to the FEA based coupling analysis described above, another type of coupling analysis was performed to look at manufacturing variations.

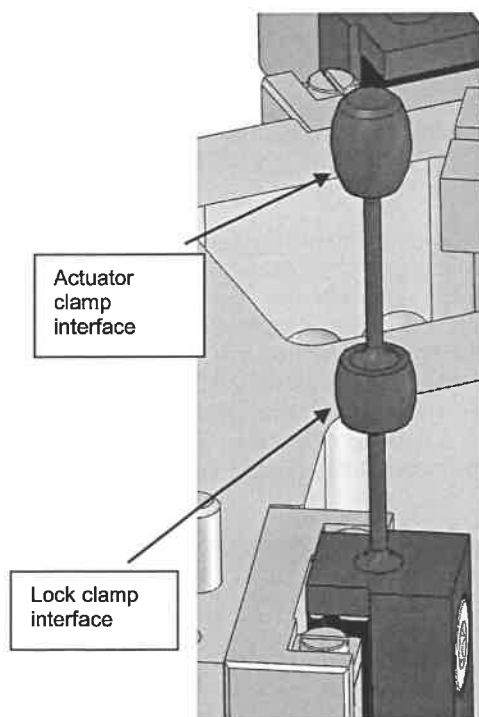


FIGURE 6. A wire-type flexural rod that couples each actuator to its hexapod leg. The upper barrel-shaped clamping interface is for actuator coupling while the lower interface is for the locking mechanism.

Because of the complex geometry of the hexapod, it is not convenient for an operator to individually adjust a single actuator to achieve a desired adjustment of the hexapod platform.

Rather, it is necessary to use a kinematic algorithm (which is contained within the hexapod controller) to convert desired displacements of the hexapod platform into the six actuator displacements. A mathematical coupling analysis was performed in MATLAB to verify that no calibration of the hexapod geometry would be needed. Because of normal fabrication and assembly tolerance variations, the as-built geometry of the hexapod will deviate slightly from the kinematic model. For simulation purposes, relatively large geometric errors were applied to the kinematic model. Normally distributed three sigma variations of 0.3 mm were randomly applied to all geometric parameters. The resulting imperfect kinematic model was then compared to a perfect model. This comparison showed that coupling of less than 1% occurred for commanded motions. (In this context coupling is defined as undesired motion in an orthogonal axis. For example, 1% coupling between X and Y motion would indicate that for a 10 micron adjustment in X and error motion in Y of 0.1 micron might be produced.) A large number of simulations were performed for this analysis.

CONCLUSION

This paper describes a high-precision hexapod with several design features that maximize positioning accuracy and stiffness over the range of motion. The approach is ideal for opto-mechanical devices that require occasional changes in position at reasonable cost.

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NANOCOINING OF OPTICAL FEATURES

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INTRODUCTION

The use of non-reflective surfaces in consumer and commercial applications has become increasingly popular. Whether they are used to reduce reflection on a monitor or increase the efficiency of photovoltaic solar cells, the advantages are being recognized and the demand is increasing. The limitation of these non-reflective surfaces is the length of time required to produce a useable quantity, usually by deposition and lithography techniques. This paper describes an approach to creating non-reflective short wavelength optical features with a high throughput.

The goal is to create non-reflective surfaces through a process called nanocoining. Nanocoining uses a nanostructured diamond die to imprint a structure on to a mold surface. The nano-scale structures on the surface counteract the large discontinuous refractive index change at air-substrate interfaces [1]. This type of manufacturing requires the design of an ultrasonic actuator (50 kHz) to produce the indentations at a high rate.

The physical process of nano-indentation is investigated to understand how material behaves at the nano-scale. A slow speed functional system has been produced by creating experimental indents and comparing them to FEA results. The ultrasonic actuator along with the ability to create nanostructured features are the key elements to producing optical quality non-reflective surfaces efficiently.

INDENTATIONS

The objective of this research is to produce a surface that resembles the outside layer of a moth's eye. Examined under magnification, the surface of a moth's eye is covered with hemispheres whose diameter and spacing is less than the wavelength of light. It is this property that creates a non-reflective surface. This research utilizes a diamond indenter with these non-reflective features on its surface. The indenter is pressed into the surface of a suitable

material (plated Cu, 1100 Al, ...) to replicate the moth's eye nanostructure.

Indent Simulations

Finite Element Analysis (FEA) was performed to model the behavior of material at the nano-scale. An indenter that would produce moth's eye-like hemispherical features was used in the FEA simulations. The modeled hemispheres had 100nm radii and 300nm spacing between centers. An image of the FEA simulation results can be seen in Figure 1.

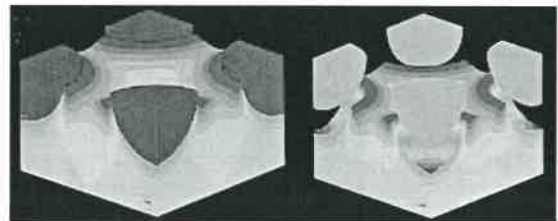


Figure 1. FEA simulation of nanocoining using 100 nm radius hemispheres.

Slow Speed Experimental Indentations

Minnesota Mining and Manufacturing Company (3M) provided a flat indenter to develop an indentation process before a nano-structured die was used. The flat indenter has a 100 x 100 μm flat face. A PZT fast tool servo (FTS) mounted to a precision 4-axis milling machine was used to create a 10 x 10 array of indents. A portion of one of the arrays on plated copper can be seen in Figure 2.

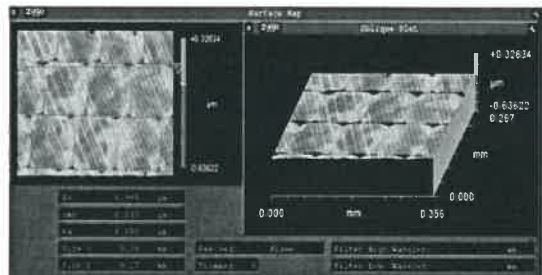


Figure 2. Portion of array created with 100 x 100 μm flat indenter mounted on FTS.

Figure 2 shows that indents created with a diamond die can be spaced, aligned and indented precisely.

Experimental indentations were also made with a nano-structured indenter from 3M. The nano-structured die is composed of an inner square grid of 1600 (40 x 40) nano-features (200 nm x 200 nm x 250 nm tall) spaced 300 nm apart. An outer border, which is at the same height as the nano-features, produces a reflective surface for alignment. An SEM image of these nano-features can be seen in Figure 3.

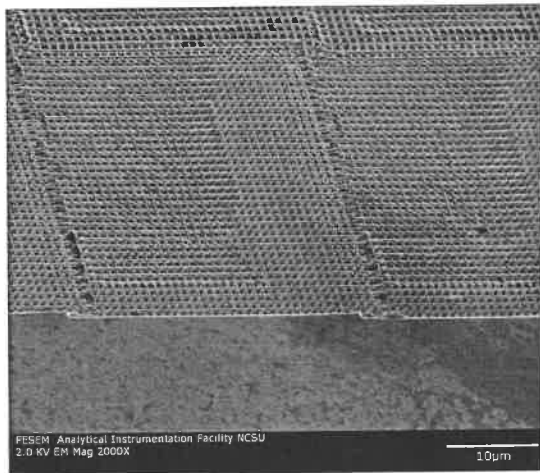


Figure 3. SEM images of multiple nano-featured indents at 2000x. Un-indented material is seen in lower region of image.

The indented material is plated copper and was indented at a frequency of 2 Hz. Edges between successive indents are also seen in Figure 3 indicating the ability to precisely index indents in both directions. A higher magnification image of the indents is seen in Figure 4. The individual nanometer-scale features are easily recognized showing that the quality of the features can be maintained even though they are being replicated at high speed.

FEA simulations coupled with experimental indentations have revealed how the material will behave during the indentation process and that nanocoining is a viable option for replication of nano-features.

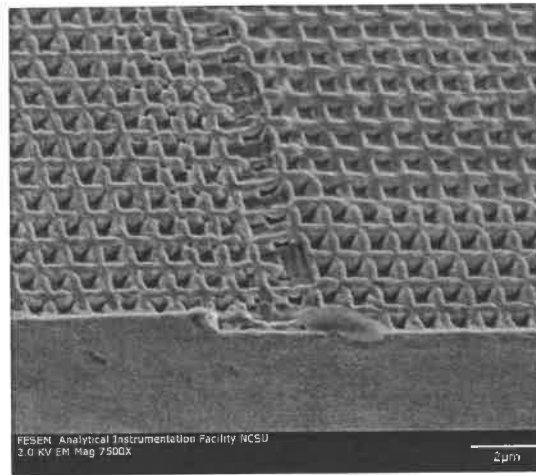


Figure 4. SEM image of nano-featured indents at 7500x. Individual imprints from nano-structures distinguishable.

High Speed Experimental Indents

The previously described indents were all created using relatively slow indentation speeds. The frequency ranged from 1 to 10 Hz. At these speeds the raster speed of the indenter across the work-piece is slow enough not to cause problems. At frequencies of 1000 Hz or more, the axis of the Nanoform can no longer keep up with an acceptable error. Therefore the tool was held stationary and the part was turned on a precision spindle. An image of this set-up can be seen in Figure 5.

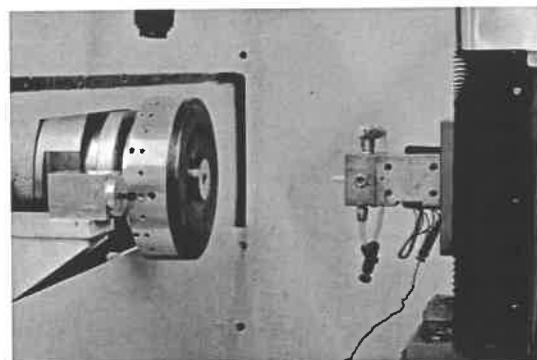


Figure 5. Experimental set-up for high speed indenting. Ultramill (right) indents while copper work-piece rotates on spindle (left).

High speed experiments were performed using the Ultramill elliptical vibration assisted machining (EVAM) actuator. The Ultramill is a non-resonant actuator normally used for EVAM experiments. The operating frequency of the

Ultramill is between 1000 and 4000 Hz. The Ultramill is actuated using two piezoelectric stacks driven 90 degrees out of phase (phase may be altered to change the shape of the tool path). An image of this concept is shown in Figure 6.

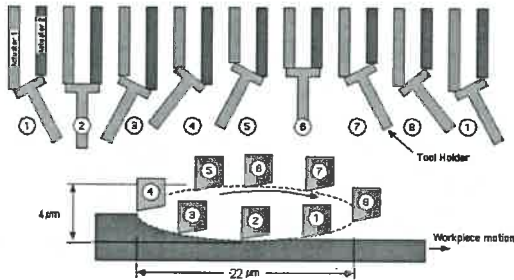


Figure 6. Description of the Ultramill's elliptical tool path generated by two piezo stacks driven out of phase.

Relationships between the rotational speed of the spindle, radial position of indenter, actuation frequency and phase angle of Ultramill drive signal can be determined so that indents are indexed properly and feed in a spiral pattern as in single point diamond turning. When all parameters have been accounted for, part velocity and indenter velocity match creating large arrays of un-deformed indents at a high rate.

Figure 7 shows individual nano-features created using the Ultramill's elliptical tool path. The part is moving from right to left in the images implying the first image shows the tool moving too slow with respect to the part, the second shows the tool moving too fast and the third shows the part velocity matching the horizontal tool velocity. This can be determined by the presence of smear and its direction.

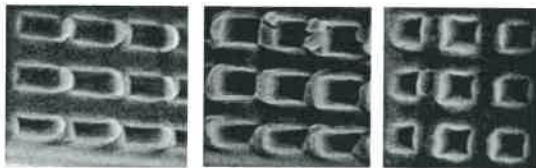


Figure 7. Nano-features created with Ultramill elliptical actuator. Left shows tool too slow, center shows tool too fast and right shows tool matching part velocity.

ELLIPTICAL TOOL PATH

Arguably the most challenging aspect of the nanocoining process is achieving an appropriate

tool path. Linear indenting (die moving into and out of the work-piece orthogonal to its surface) would work if the work-piece is stopped every time the die comes into contact. This, however, is not possible due to the high frequency requirement. Since the work-piece must be continuously rotating, plunging the die into a moving work-piece would cause the die to drag along the surface creating smear. To eliminate the smear effect the velocity of the die must match the velocity of the work-piece at the time of contact. An elliptical tool path would allow tool motion orthogonal to the work-piece (indent direction) as well as along its surface allowing the die to match velocities and index back to the next indent location. A description of the elliptical tool path is shown in Figure 8.

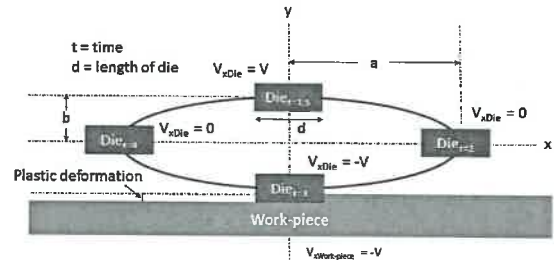


Figure 8. Description of elliptical die path and velocities for one cycle.

A MATLAB simulation of the indentation process was written to define appropriate values of the 'a' and 'b' dimensions of the ellipse (Figure 8). The outcome of developing the simulation was that the 'a' dimension is a function of the die length 'd' only. The 'b' dimension is an independent parameter. One function of the simulation was to calculate the amount of smear for different values of 'a' and 'b.' The amount of smear is quantified as a percentage of velocity difference. Initially, it was assumed that a more horizontal ellipse shape ('a' larger than 'b') would have the best characteristics. However, after iterations trying to reduce the smear, the most attractive shape was vertical ('b' larger than 'a'). A vertical ellipse reduces smear by decreasing the time in contact between die and work-piece. The dimensions of the ellipse for a 50 kHz actuation frequency were determined to be $a = 4.5 \mu\text{m}$ and $b = 8.0 \mu\text{m}$.

ACTUATOR

An ultrasonic actuator is required to have a realistic system capable of producing large scale quantities of non-reflective surfaces. The dimensions of the indenter are on the order of

20 to 50 μm (side length of square indenter) so a target frequency of 50 kHz was chosen. Such high frequencies suggest the use of a piezoelectric material as an actuator. Also, since significant displacements are desired (relative to 50 kHz) the actuator must operate at resonance. One technology that fits these criteria is the Bolted Langevin Transducer (BLT).

A BLT is an actuator that consists of piezoelectric material sandwiched between two masses. A bolt runs through the piezoelectric material and clamps the two masses together creating an arrangement which is essentially a spring-mass system. The operating frequency is thus based on the mass and stiffness of the system. A 'horn' (some sort of truncated rod) is fit on the end of the BLT and tuned to have mode shapes at the operating frequency of the BLT. The horn then acts like a mechanical amplifier (amplification is a function of truncation). The difficulty of implementing a sonotrode (BLT fitted with horn) is obtaining displacement in two directions. Most commercial BLTs have piezoelectric material poled so that displacement is along the length of the sonotrode. Adding an imbalanced mass will cause deflection in a second direction, but the phase difference will always be 180 degrees instead of the desired 90 degrees. This implies that either multiple sonotrodes would be needed (for multiple actuation directions) or the piezoelectric material would have to be poled in a way that would generate multi-directional displacement.

Another type of actuator under consideration is an active structure composed entirely of piezoelectric material. The structure would be designed to have two resonant modes at 50 kHz. Each mode would contribute to one direction of actuation. The modes would then be excited by actuating the appropriate part of the structure. The PEC has had success with active, multi-mode resonant structures in the past. One such structure, shown in Figure 9, was designed to achieve elliptical motion used to rotate a drum.

The actuator shown in Figure 9 is composed of two layers of piezoelectric material, each layer having three electrodes. A longitudinal mode is excited by actuating both center segments of material in phase. A bending mode was activated by exciting the outer electrodes in

phase (one layer opposite voltage to achieve bending). The longitudinal and bending modes were actuated out of phase to obtain elliptical motion. Different geometries more applicable to the nanocoining tool path have been designed, but a structure utilizing the same principles as those used in the actuator in Figure 9 could be a possible solution.



Figure 9. Active multi-mode resonant structure used to generate elliptical motion for rotation of drums.

CONCLUSIONS

The use of a diamond indenter to nanocoine optical features could be an efficient way to produce large scale quantities of non-reflective surfaces. Simulations have been compared with slow speed experimental indents to determine the optimal method of producing repeatable, nano-structured indents. Experimental indentation procedures currently developed have been successful and are continuously increasing in production rate. An ultrasonic actuator will be developed to generate an elliptical indenter path at 50 kHz resulting in optical quality nano-features.

ACKNOWLEDGEMENT

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3-D Metrology and Correction of Aspheric Diffractive Optics

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Overview

Historically, it has been challenging to measure diffractive optical surfaces and deterministically correct for symmetric and asymmetric errors inherent in the turning process. The diffractive zones (for IR applications) tend to fall between 3 microns and 25 microns in depth, depending on the refractive index of the material. Compared to common form error specifications of 0.150 microns to 0.633 microns, the large diffractive zones hide the true form error and make it challenging to measure, let alone correct for form error.

Process

Taylor Hobson and Precitech have developed a process to measure, analyze and correct for aspheric and diffractive form errors. This process utilizes Taylor Hobson's PGI 3-D with Advanced Aspherics Analysis (AAU) software, along with Taylor Hobson's CCI (Optical Profiler) and Precitech's NanoForm 250Ultra with a rotary B axis and a Fast Tool Servo (FTS).

Spherical Surface Correction

The measurement and manufacturing process will first be demonstrated by cutting and measuring a spherical shape into a sample. The measurement of radius and form error will be done with the PGI 3-D as well as an interferometer. These results will be compared, and the NanoForm will be used to correct the measured form error.

Aspherical Surface Correction

The measurement and manufacturing process will secondly be demonstrated by cutting and measuring an aspheric shape into a sample. The measurement of absolute radius and form error will be done with the PGI 3-D, followed by a correction cycle with the NanoForm.

Diffractive Surface Correction

Finally, the measurement and manufacturing process will be demonstrated by cutting and measuring an aspheric diffractive shape into another sample. The rotary B axis will be utilized in conjunction with a split radius tool to machine the diffractive steps. It will be shown that when used in tool normal mode, a split radius tool can rapidly machine an excellent surface and a highly efficient diffractive structure. The measurement of radius and form error will be done with the PGI 3-D. A detailed measurement of the diffractive zone depth, corner radii, and surface finish will be made with the CCI. These measurement data will then be used by the NanoForm to reduce the form error in three dimensions using the Fast Tool Servo.

MICROFABRICATION OF TERAHERTZ VACUUM ELECTRONIC DEVICES

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INTRODUCTION

High resolution imaging, and high data rate communication/sensing systems have long necessitated the development of high power RF vacuum electronic devices (VED) that operate in the terahertz region (0.1–10 THz) [1]. In particular, visual impairments such as fog or rain have obvious impacts on observations in the visible spectrum. While millimeter waves exhibit significant attenuation owing to their immense water absorption, there exist “windows”, such as one at 0.2 – 0.3 THz that can be exploited to penetrate these obscurities [2]. Aircraft could benefit from all weather capabilities by using this frequency range to look through fog, rain, and other water based atmospheric conditions [2]. Active and passive THz imagers could be of immense importance in a variety of applications, but there are several challenges to overcome if their utility is to be realized. Passive imaging systems so far have been bulky with poor performance, while active systems have been limited by an available THz power source [2].

To further develop active THz imaging, broadband sources of THz waves are essential. However, conventional electronic technology encounters a technical limit in the THz regime [3], so we have developed a novel VED as the ultra wideband THz source. Our approach to this problem involves the construction of a traveling wave tube (TWT). TWTs were first conceived more than 60 years ago by Rudolf Kompfner [4]; however, the fabrication of a TWT in the terahertz region presents significant manufacturing challenges [5]. Devices that operate at these frequencies have microstructures on the order of tens to a few hundred microns, and tolerances of a few microns. There are many approaches for the microfabrication of these devices, such as Electric Discharge Machining (EDM), Deep Reactive Ion Etching (DRIE), and lithography methods (LIGA) [5].

However, one technology that has been largely overlooked is CNC milling. The common wisdom has been that direct machining cannot meet the sensitive tolerances of high frequency TWT circuits (dimensional tolerance < 3 μm , surface finish < 100 nm R_a) [6].

However, since the invention of the Mori Seiki NN1000, we have successfully fabricated broadband (~60 GHz), 0.22 THz TWT circuits [6]. In this paper, we will discuss the basic operation of TWT circuits, and why they have extremely sensitive manufacturing tolerances, the capabilities of the NN1000, the machining techniques employed, and we will conclude with a summary of our findings.

TRAVELING WAVE TUBE (TWT)

Our goal is to create a broadband vacuum electronic (VE) high-power amplifier (HPA). There are three major constituents for our device, an electron gun, interaction circuit, and collector. The electron gun will emit a high current density (>100 A/m²) sheet beam that will pass through the TWT interaction circuit. Inside the TWT, energy from the electron beam will be converted to terahertz radiation. After the beam emerges from the interaction circuit it will be allowed to disperse in the collector [7]. This paper focuses on the fabrication of the TWT. Our circuit is designed to operate at a central frequency of 0.22 THz [1]. We use a half-staggered double grating array, to amplify THz signal via interaction with the free electron beam. Figure 1 shows a cross-sectional view of the circuit concept. Oxygen-free copper is a good material choice due to its high conductivity ($\sigma = 5.88 \times 10^7 \Omega^{-1}\text{m}^{-1}$); however, there are some difficulties with machining, as discussed below. The microstructure of the grating has a period of 460 μm , consisting of alternating vanes and pockets measuring 115 and 345 μm respectively, and an overall length of 40 mm [1].

Simulations show that the resulting electromagnetic (EM) interaction will generate 0.22 THz radiation with increasing strength of 12 dB/cm over 25% bandwidth [7]. Similar devices exist for RF and microwave frequencies, but millimeter and sub-millimeter wave devices have prominent concerns with energy dissipation on lossy metal surfaces [7].

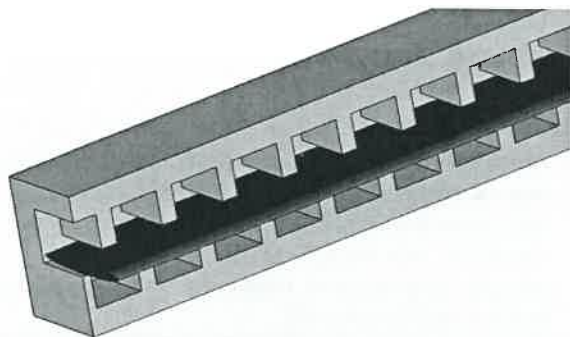


FIGURE 1. Circuit concept showing electron beam (center) passing through the half-staggered grating.

ULTRA-PRECISION MACHINE

The Mori Seiki NN1000 is a 5-axis (X, Y, Z, A, and C) CNC machine developed by Digital Technology Laboratory (DTL, a subsidiary of Mori Seiki) in Davis, CA. It features air-bearing guideways, vibration isolation technology, and laser scale position sensors. The laser scales have a resolution of 34 pm, and work in concert with linear drive motors to allow position control resolution of 1 nm. Cartesian travel limits allow for work pieces of up to 150x120x50 mm. The machine can use either an air turbine spindle capable of 55,000 RPM or a fixed mandrel for scribing process [8].

The NN1000 is a smooth and steady platform designed for ultra-precision work with exceptionally fine surface finishes. The capabilities of the machine depend on the details of the work, such as materials, techniques, and tooling. Using single-crystal diamond milling and scribing tools, mirror surfaces can be machined without polishing (<10 nm R_a).

MACHINING PROCESS

Careful selection of materials, tools, and cutting strategy are critical for success. We used standard tungsten-carbide tools with AlTiN coating, with diameters ranging from 102 to 1,270 μm and cutting length to diameter ratios of 3:1 to machine the 0.22 THz TWT circuit. We were able to achieve tolerances finer than 1 μm

and surface finishes of 40 nm R_a . Minimum surface finish requirements are related to the EM skin depth of the material for the frequencies of interest [9]. For 0.22 THz in copper, the skin depth is approximately 140 nm. Our tests have shown significant improvements in signal transmission as surface finishes improve to ~100 nm R_a , and diminishing returns thereafter. Problems with burr formation are a serious issue because even small burrs measuring 5 or 10 μm are large compared to circuit dimensions. Since circuit features are small and fragile, we are very limited on what types of post processing can be done to reduce burrs and improve surface finish. We use a mild acid bath to clean and treat the parts after machining. Careful consideration must be given to acid washing, since excessive etching will reduce the surface quality.

Micromachining copper proved problematic. The ductility of copper leads to large burr formation if feed rates are too slow, or if the tooling becomes dull [10]. Many of our circuit features are sufficiently small that the machine never has sufficient space to accelerate to ideal feed rates. The result is underfed tools that “plow”, rather than cut the material [10]. This also causes increased tool wear which exacerbates this problem. Employing dispersion-strengthened copper, such as Glidcop, greatly improved part quality. Glidcop is copper with ultra-fine alumina particles. The alumina serves to reduce the grain size of the material and increase its structural rigidity [11]. We found that using Glidcop greatly reduced burrs and increased surface quality in hard-to-reach places. The alumina contributes to tool wear, and Glidcop has about 8% less conductivity than oxygen-free copper, but these issues are offset by increased part quality.

Machining techniques further minimize burr formation. When a new feature is to be machined we first cut the material at a depth of one micron, and we leave a stock allowance of ten microns from the edge of any feature. Repeated machine passes will cut the material to within one micron of its desired dimensions, before performing a finishing pass. This process is repeated with gradually increasing depth, until there is enough material support to resist burr formation, typically 10 to 15 microns deep. Once the features are deep enough, more aggressive machining takes place to reduce cycle times. Similar techniques are used to finish the floor of any feature, to get the correct depth to within 1 micron, and to achieve the highest possible surface quality.

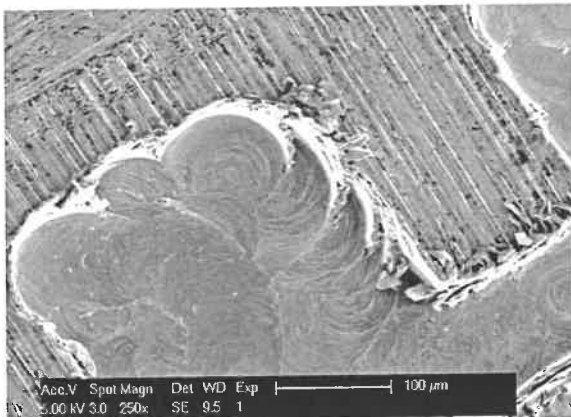


FIGURE 2. SEM image of micro-machined Cu.

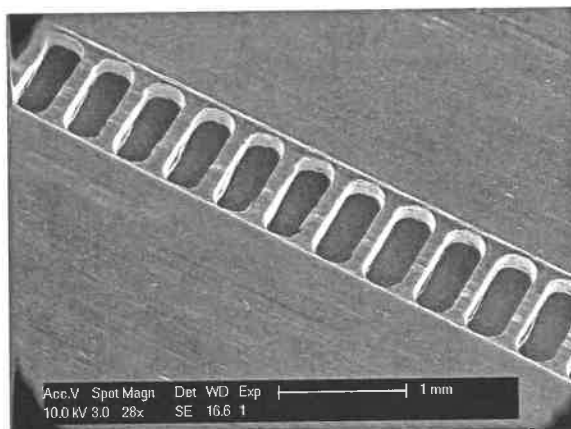


FIGURE 3. SEM image of interaction circuit made from Glidcop.

We found that surface speeds of 40 to 80 m/min and chip loads of 0.7 to 3 μm per tooth provide a good balance between performance and wear. Cutting depths vary from 2 to 10 μm per pass depending on the tool size and type of cut. With these parameters, we found that tool length wears ~ 50 μm per hour, with an average life expectancy of 2 hours. Radial tool wear was negligible except on the corners. A corner radius of several μm develops as soon as machining is started. A few micron radius in the corners is tolerable, but the axial cutting edge must be monitored in order to cut correct feature depths and maintain good surfaces. If chips are allowed to accumulate on the tooling, then tools less than 250 μm diameter are prone to bind and break. Our circuits have aspect ratios up to 4:1, which makes chip evacuation a concern, and micro-pocket features are excellent at capturing chips. Cutting fluids proved to be counter-productive; micro-chips become suspended in the fluid, acting like a lapping compound and reduce tool life to a few minutes. Chip removal

was accomplished by injecting an air stream around the cutting tip.

Using the largest tool that will fit into the part minimizes tool wear and breakage problems. We split our TWT circuit along the major axis of the rectangular beam tunnel cross-section. This allowed us to double our tool diameters for pocketing operations from 127 to 254 μm . However, this placed a seam in the finished part down the center of the magnetic field maximum, thereby disrupting the surface currents and EM propagation. In this case, the TWT circuit will have good performance only if the parts are perfectly aligned and well bonded. Fortunately, the NN1000 excelled in overcoming this problem. We integrated rectangular alignment registers with a small interference fit. We also diffusion bonded our circuit halves together, providing the necessary conductivity and sealing the circuit cavity for vacuum. Since diffusion bonding involves heating the parts up to 1000 $^{\circ}\text{C}$, Glidcop has an additional advantage over copper. It is as much as ten times stronger than copper after annealing [12], and the parts are less likely to distort under high temperature and pressure.

CONCLUSION

The final evaluation of our TWT circuit involves an elaborate and time consuming process, however, there is a simpler way to test our work during the development stages. We can omit the electron beam and send low-intensity EM radiation through the circuit. While varying the input frequency and monitoring the input and output power, we can measure the frequency response of the system. This is an efficient evaluation that we have dubbed the "cold test".

Computer simulations predicted that our circuit should attenuate all frequencies below 0.20 THz, above which the signal should rise abruptly, and have good signal transmission until about 0.28 THz. We measure the output transmission loss (S_{21}) as the log of the ratio of output over input intensities. With ideal conditions we expect to see S_{21} at minus 3-5 dB with about 70 GHz bandwidth. Improving machining techniques has steadily improved our results. Our most recent cold test correlated closely with simulations. S_{21} transmission had about 3.5 dB loss over a bandwidth of 60 GHz, figure 4. Total cycle time is about 40 hours for a single circuit. While tungsten carbide tooling is inexpensive, they do wear rapidly in this application. Optimizing machining conditions

and tool selection is continuously in development.

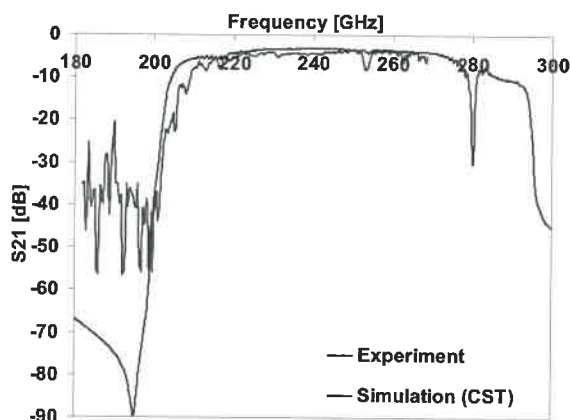


FIGURE 4. Comparison of simulated power transmission and cold test results. Experimental results end at 270 GHz due to the limitation of our EM source.

"Nano-milling" is extremely versatile. CNC programs can be written in a few minutes, and after a few hours or days, depending on the complexity of the part, researchers can be testing their designs. High aspect ratio parts can be made with vertical walls [8]. 5-axis capabilities allow for exceptionally complex shapes. Other techniques can easily take several months of development, and many are restricted to essentially a 2D approach.

We would like to acknowledge DTL for allowing us generous access to the NN1000, and the collaboration of DTL engineers. Funding and project support is provided by Teledyne Scientific through the DARPA HiFIVE program.

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IMPROVED SURFACE FINISH IN STEEL DESPITE EXCESSIVE TOOL WEAR USING ELLIPTICAL VIBRATION ASSISTED MACHINING

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INTRODUCTION

Multiple researchers have shown improved surface finish and tool wear when utilizing elliptical vibration-assisted machining (EVAM) [1]. EVAM applies micrometer-scale elliptical vibration to a diamond tool. EVAM can produce optical-quality surface finishes on steel, which otherwise causes drastic wear to diamond tools during conventional (non-EVAM) diamond turning. This high level of wear is attributed to a thermally activated chemical reaction between the diamond and iron-based alloy. Wear of diamond tools cutting steel have shown to follow an Arrhenius-type relation with tool temperatures obtained from finite element modeling [2]. This model assumes the following form:

$$\frac{dV}{dt} = A \exp\left(\frac{-E_a}{RT}\right) \quad (1)$$

where V is the worn volume (cross-sectional wear area $\times$ width of cut), R is the gas constant (8.3144 J/mol), and T is the peak tool temperature (K). A and E_a , the pre-exponential constant and activation energy, are empirical constants specific to the interacting materials (diamond and AISI1215 steel in this case). These values were previously determined to be $4.5 \times 10^7 \mu\text{m}^3/\text{s}$ and 42.5 kJ/mol, respectively [2]. Further FE simulations based on those used to develop the Arrhenius model using AdvantEdge by Third Wave Systems were run that incorporate a wider range of velocity and depth of cut values. These were used to develop a relationship between these parameters and tool temperature, the results of which are shown in Figure 1. When this relationship is supplied to the Arrhenius wear model in Equation 1, it forms a relationship between wear rate and workpiece speed. Dividing dV/dt in Equation 1 by workpiece speed gives wear per cutting distance, with depth of cut and workpiece speed as variables. This relationship is shown in Figure 2. Other researchers have observed

similar relationships with wear and machining speed, but did not relate this to an Arrhenius-type chemical reaction [3].

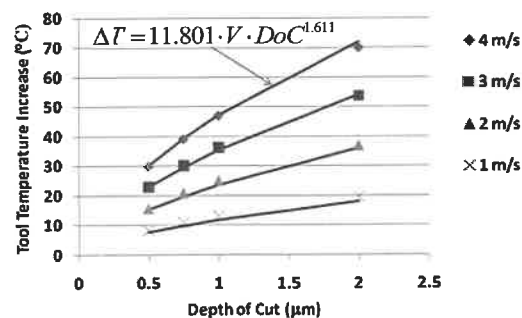


FIGURE 1. Relationship between tool temperature rise above ambient (20 °C), depth of cut, and velocity for single crystal diamond tool on low carbon steel determined from AdvantEdge FE simulations.

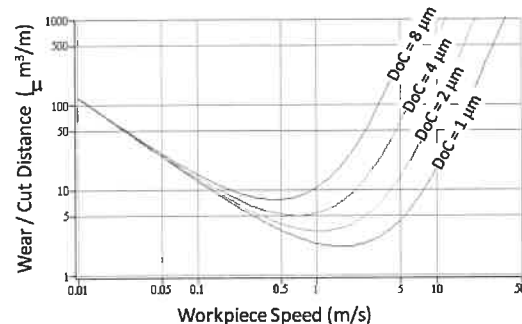


FIGURE 2. Arrhenius tool wear model calculated from wear experiments and FE tool temperatures of non-vibration-assisted diamond turning of AISI1215 steel.

Despite various claims of reduced wear, few have measured the worn cross-section of the tool after EVAM, but rather note wear land length from a microscope perspective. Using a unique method called electron beam-induced deposition (EBID, [4]) tool wear was measured

after EVAM of AISI1215 steel. Along with tool wear, surface finish was measured. EVAM tool wear and surface finish were compared to those obtained after conventional machining of the same material. Wear results from the EVAM experiment can be explained by the Arrhenius model in Figure 2, though the surface finish proved to be a unique result of the EVAM process and not necessarily the tool wear.

EVAM OF STEEL

The relationship between frequency and tool wear were tested with a sub-ultrasonic EVAM tool called the Ultramill available at the Precision Engineering Center at NCSU [1]. An AISI1215 steel workpiece was fabricated that allowed specific regions of wear on a straight-edged, single crystal synthetic diamond tool provided by Chardon Tool. This allowed wear to be measured after all cutting was complete. Figure 3 shows the experiment setup. Cutting experiments were conducted on a Rank Pneumo Nanoform 600 diamond turning machine. Mobilment Omicron cutting oil was used as lubricant in all cutting tests.

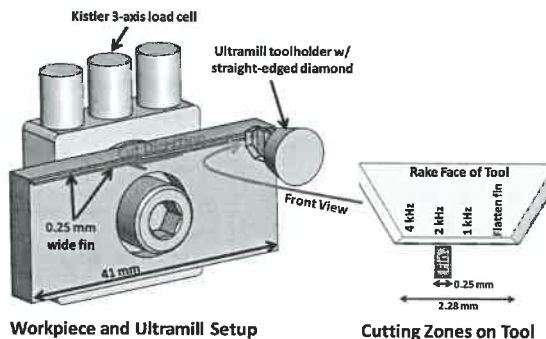


FIGURE 3. Schematic of EVAM experiment setup and localized wear zones on the diamond tool.

Workpiece speeds with EVAM are limited by the surface finish produced by the oscillating tool. This is determined similarly to predicted surface finish due to tool radius and cross-feed in conventional single point diamond turning, but uses ellipse major radius and upfeed per vibration cycle. Upfeed surface finish is determined by the following:

$$PV_{up} \approx \frac{b}{8} \left(\frac{v_{wkpdc}}{a} \right)^2 \quad \text{for } \frac{v}{a\omega} \ll 1 \quad (2)$$

where a is horizontal (parallel to workpiece surface) ellipse amplitude, b is vertical ellipse amplitude, f is EVAM frequency, and v_{wkpdc} is workpiece velocity. Depth of cut was set to 2 μm , and tool tip vibration amplitude (1/2 total ellipse width) was set to $axb = 11 \times 2 \mu\text{m}$. Workpiece velocity was increased with tool frequency in the EVAM experiments to maintain a constant chip thickness and theoretical surface finish of 9nm PV. Table 1 gives the frequencies and workpiece speeds used.

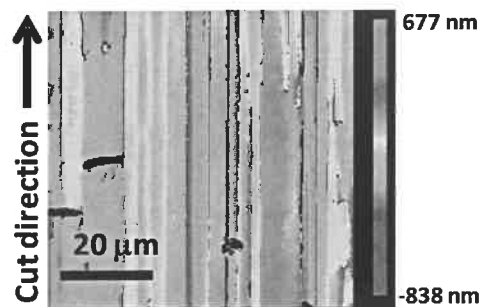
TABLE 1. EVAM parameters varied during tool wear experiments.

Cut #	Tool Frequency (Hz)	Workpiece Speed (mm/s)	Machined Distance (m)
1	1000	2	5
2	2000	4	5
3	4000	8	5

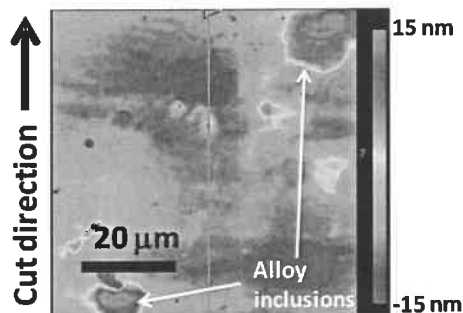
Since velocity plays such an important role in determining wear rate, and EVAM experiments are limited to very low workpiece speeds, conventional machining experiments were also conducted at low speeds without elliptical tool vibration. The same workpiece geometry, material, and workpiece speeds shown in Figure 3 and Table 1 were used in non-EVAM cutting experiments for comparison. Also, results from a previous experiment at high workpiece speed (1 m/s) is given for comparison [2].

RESULTS

The EVAM experiments produced optical quality surface finish (<10 nm Ra, <15 nm rms) on AISI 1215 steel surfaces which did not vary with frequency. Conventional DT experiments on the same steel produced poor surface finish (>100 nm Ra and rms) for the high speed and low speed experiments. Scanning white light interferometer (SWLI) surface images in Figure 4 show the main contribution to these roughness values was dragging of hard alloy particles along the cutting direction of the workpiece surface. The EVAM surface finish showed little dragging, and in fact showed alloyed inclusions 'stuck' in stationary locations. Observation of both tools with a Nomarski microscope showed considerable amount of metal pickup on the conventional tools which was not evident on the EVAM tool. The lack of metal pickup when using EVAM was also observed by other researchers [3,5].



AISI 1215 Steel, Conventional Machining
40 m machining distance
 $R_a = 122 \text{ nm}$, $rms = 151 \text{ nm}$



AISI 1215 Steel, 4 kHz EVAM
15 m machining distance
 $R_a = 8 \text{ nm}$, $rms = 14 \text{ nm}$

FIGURE 4. Top: SWLI image of AISI 1215 steel after non-vibration machining. Bottom: SWLI image after EVAM of same material.

Both EVAM and conventional tools were measured for wear using the EBID method. Wear profiles for the EVAM tool varied little between the 1 kHz and 4 kHz experiments therefore are not depicted in Figure 5. Worn tool tip cross-sections shown in Figure 5 for both machining methods showed a greater level of wear using EVAM at shorter distance compared to high speed machining. This can be attributed to the low tool tip velocities available in sub-ultrasonic EVAM ($<0.28 \text{ m/s}$), which occur below the wear minimum shown in Figure 2. Also, the reduced EVAM chip thickness, similar to depth of cut in conventional machining, contributes little to the wear in this region. The low speed, non-vibration experiments yielded an even higher level of wear at the same machining distance than the EVAM experiments. This shows, that EVAM does in fact reduce tool wear, but only when compared against cutting at the

same workpiece velocity. Excellent surface finish, at least in the cutting direction, can be obtained using EVAM even with a substantially worn tool.

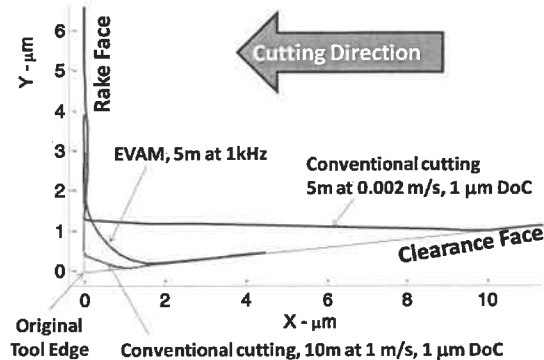


FIGURE 5. Worn tool cross sections from EVAM and conventional machining of steel measured using EBID method [4].

DISCUSSION

Speed has been shown to play a dramatic role in the thermo-chemical wear of diamond. The wear rate is exacerbated by tool temperature, and temperature increases with workpiece velocity. However, low speed machining requires longer contact time to cover the same distance, and this contact time allows the tool to wear more over the same distance. The low workpiece speed and tool tip velocities of EVAM gave higher wear rates when compared to non-EVAM machining at 1000x higher speed. However, this wear was far less than that measured during non-EVAM machining at the same EVAM workpiece velocities. This shows that EVAM does reduce tool wear, but only when compared against cutting at the same workpiece velocity.

Improved surface finish with EVAM is largely due to the unique machining process and not reduced tool wear. This is attributable to the absence of workpiece material pickup on the EVAM tool and no particle dragging in the produced surface, whereas non-EVAM cutting had material pickup on the tool and drag marks in the workpiece surface. No pickup was also observed on the EVAM tool in [5]. It is hypothesized that motion of the tool in EVAM allows it to pass over and embed alloy inclusions into the workpiece. It is also hypothesized that any built up edge on the EVAM tool is removed during subsequent elliptical passes.

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5-Axes Grinding of Micro-Patterns with Adjusted Grinding Wheels

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INTRODUCTION

Friction significantly reduces the effectiveness of gas turbines. Micro roof profiles (riblets) can theoretically reduce the near wall friction up to 10 %. Therefore, these structures have to be oriented in stream direction with an aspect ratio of riblet-width to riblet-height of 0.5 [1, 2]. For the technical application of riblet-structures on compressor blades, riblet-heights between 20 μm and 120 μm are required. So far riblet-structures with aspect ratio of 0.5 and a width of 30 μm have been produced on NACA6510 profiles by grinding and reduced the near wall friction about 4 % [3]. In contrast to this profile, real compressor blades have double curved or even free formed surfaces. In case of grinding of flat or single curved surfaces the macro geometry of the grinding wheel is equal to the shape of the work piece. Due to the changing curvature of a free formed surface the macro geometry of the grinding wheel differs from the shape of the workpiece [4]. As a result, the shape of the compressor blade will be changed. But the deviation of the compressor blade can influence the performance negatively.

Next the micro profiles have to be dressed on the grinding wheel. Due to the high profile wear at application of vitrified bonded grinding wheels, riblet geometries with an aspect ratio of 0.5 are limited to a riblet-width of 60 μm [3]. Here, metal bonded grinding wheels provide a favorable wear behavior, but they are difficult to dress mechanically. Latest investigations show that electro contact discharge dressing (ECDD) is in principle suitable to generate complex profiles on metal bonded grinding wheels [5, 6]. By applying ECDD dressed tool with a single profile a significant increase in the achievable aspect ratio of the riblet-structures is realizable [3]. However, the manufacture of riblet-structures with single profile grinding wheels is not efficient. The use of the adjusted and dressed grinding wheels is the last step in the manufacturing process of riblet-structures on compressor blades [Fig. 1]. Grinding of riblet geometries on free formed surfaces require a five axis process. Here, the grinding wheel is tilt and rotated relatively to the workpiece. Due to the contact

length, undercuts can change the ground riblet-geometries. Nevertheless, the influence of the five axes kinematic on the riblet-structures is not investigated until now.

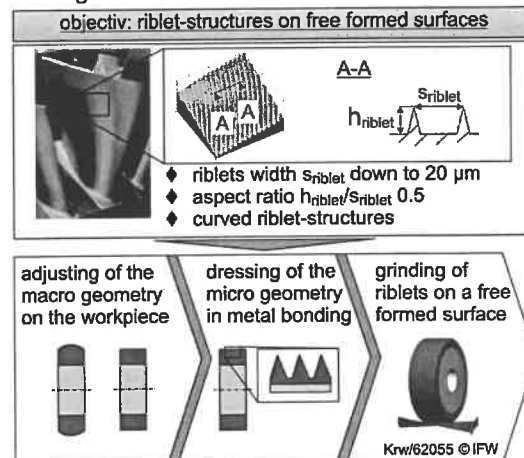


FIGURE 1. Challenges and solution by grinding of free formed surfaces

This paper first considers the influence of the macro geometry of the grinding wheel on the workpiece shape and then looks at a strategy for dressing of micro profiles on a metal bonded grinding wheel. Finally, the impact of the five axes kinematic on the riblet structures is discussed and ground riblet geometries on a free formed surface are presented.

MACRO GEOMETRY OF THE GRINDING WHEEL

The macro geometry of a grinding wheel influences the ground workpiece shape. In order to adapt the grinding wheel on the generated surface, the correlation of macro geometry, workpiece curvature and workpiece shape has to be known. Here, the incongruence defect (S_f) is used to describe the quality of the macro geometry. S_f is the maximum deferens between target and ground surface. Whereby, a positive value corresponds to damage of the target surface. A negative S_f is equal to partial penetration and the target surface will not be reached. A negative incongruence defect is limited to the offset or to the depth of cut.

The incongruence defect was calculated dependent of the workpiece curvature for different macro geometries to identify the impact factors. Here, the workpiece curvature is defined as the inverse of the workpiece radius r_{ws} , which is orthogonal to the stream direction. Figure 2 shows the incongruence defect dependent of the width of the grinding wheel. If the width is reduced, the deviation declines, too. Thus, grinding wheels with just one profile generate the smallest possible defect. But the width influences S_f just for concave surfaces.

Beside the width, the radius r_{wz} affect the incongruence defect. Whereby, a smaller radius r_{wz} generates a smaller defect. Furthermore, r_{wz} enables a constant defect for concave and convex surfaces. But a constant, negative incongruence defect means that parts of the edges of the grinding wheel are not in engagement. The three icons in Figure 2 illustrate the engagement of a grinding wheel. The comparison of the middle icon and the left one display that the maximum defect is equal to the offset, but the width of engagement is less. During grinding of riblet structures on compressor blades, a varying engagement results in unstructured areas, which cannot be structured in a second step without damaging of existing riblets.

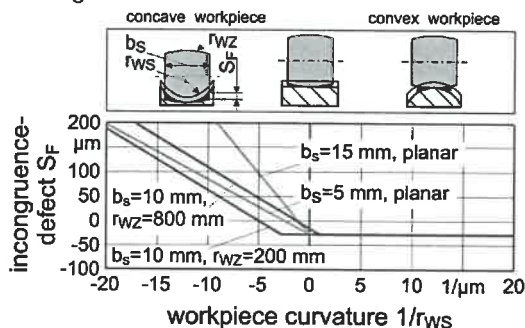


FIGURE 2. Incongruence defect for different macro geometries

MICRO GEOMETRY OF THE GRINDING WHEEL

After adjusting of the macro geometry, the micro profiles have to be dressed. Hereby riblet-geometries with a height down to 20 μm and an aspect ratio of 0.5 are required. Due to a radial infeed, the wire does not create vertical flanks in the generated roof. Therefore a shift ECDD strategy was applied [Fig. 3]. Here, the copper wire with a diameter of 1 mm is moved radial to the grinding wheel and a trapezoid roof is

dressed. In a second step the wire is displaced about Δz and moved radial again. That way several profiles were dressed on a metal bonded grinding wheel.

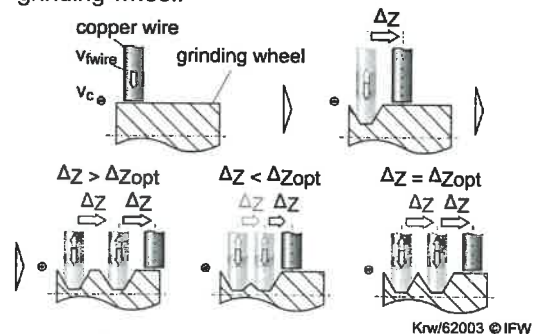


FIGURE 3. ECDD strategy

The displacement Δz has been varied to find Δz_{opt} , which produce the smallest profile geometries. Is Δz too big trapezoid profiles will be generated, but for smaller Δz the roofs overlap each other. Further investigations show that this reproducibility decrease for this case. As a result, the roofs have to touch each other at Δz_{opt} . After determination of Δz_{opt} , several profiles were dressed at the grinding wheel. The results were evaluated with the profile heights (h_{20} and h_{60}) at a profile width of 20 μm and 60 μm . In order to grind riblet structures with a riblet width of 20 μm and an aspect ratio of 0.5, the profile height h_{20} should be nearly 10 μm .

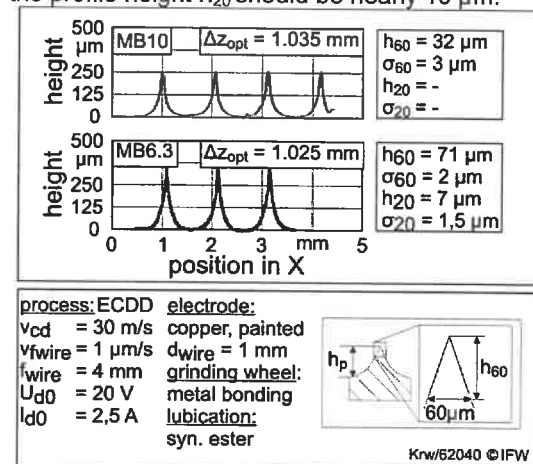


FIGURE 4. ECDD dressed micro profiles

Grinding wheels (MB6.3 and MB10) with two different grain size (6.3 μm and 10 μm) were applied in this investigation. The displacement Δz_{opt} for the grinding wheel MB10 was 1.035 μm . Here, a profile height of 32 μm at a profile width of 60 μm was reached. A profile height h_{20} could

not be measure. The minimum profiles on grinding wheels with a grain size of $6.3\ \mu\text{m}$ were at a Δz_{opt} of $1.025\ \mu\text{m}$. Compared to the grinding wheel MB10 the profile height at a width of $60\ \mu\text{m}$ increased about 100 %. The displacement Δz_{opt} is dependent on the grain size, due to the dressed roof width. During the dressing process the metal bonding smelt and grains fall out and generate a roof. If a grain with a size of $10\ \mu\text{m}$ is falling out, the width of the roof increase about $10\ \mu\text{m}$. In case of a smaller grain the enlargement is also smaller. The minimum possible profile geometry is dependent on the grain size too. On the one hand side the profile geometry is limited by the grain size, due to the fact that at the peak of the profile can be just one grain. The bigger the grain is the bigger the profile geometry. In addition, a great number of grains on the peak of a profile added stability of the peak.

RIBLET-STRUCTURES ON DOUBLE CURVED SURFACES

Due to the contact length of a grinding wheel, undercuts can emerge in the case of five axis grinding and affect the ground geometry. In order to investigate the impact of process kinematic the influence of the tool path radius on the side angle of a roof was calculated in the case of a circular tool path. For a straight tool path the side angle of the ground roof is 60° . Figure 1 show that the side angle of a roof is depended of radius of the tool path and the radius of the grinding wheel as well.

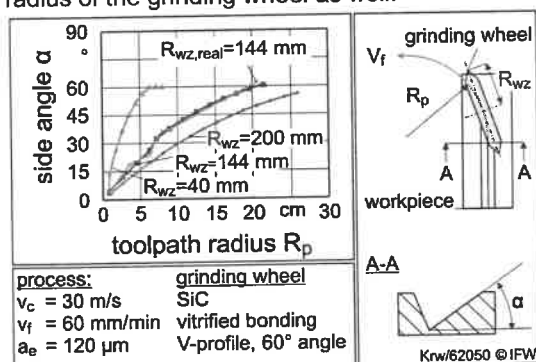


FIGURE 5. Impact of the tool path radius on the side angle

By applying a grinding wheel with a radius of $40\ \text{mm}$, the side angle is changed till a tool path radius of $6\ \text{cm}$. Whereby, the side of the roof leveled out when the tool path radius decreased. The possible tool path radius, which don't change the side angle rise which bigger grinding

wheel. For example, a grinding wheel with a radius of $200\ \text{mm}$ needs a tool path radius about $30\ \text{cm}$. The calculated results were also verified in grinding investigations with a grinding wheel radius of $144\ \text{mm}$. The comparison of calculated side angle and measured one are similar [Fig. 5].

Figure 6 displays the impact of the tilt movement of the grinding wheel. In grinding experiments a grinding wheel with a V profile starts to grind orthogonal to the plan surface and was continuously tilt to the left hand side on a grinding length of $10\ \text{mm}$. The total tilt angle was varied between 2° and 15° . The depth of cut was constant. The tilt of 2° and 4° on a length of $10\ \text{mm}$ has no impact on the side angle. A tilt angle of 10° and 15° has a great impact on the roof profile. While the right roof side did not change the side angle of the left side decrease.

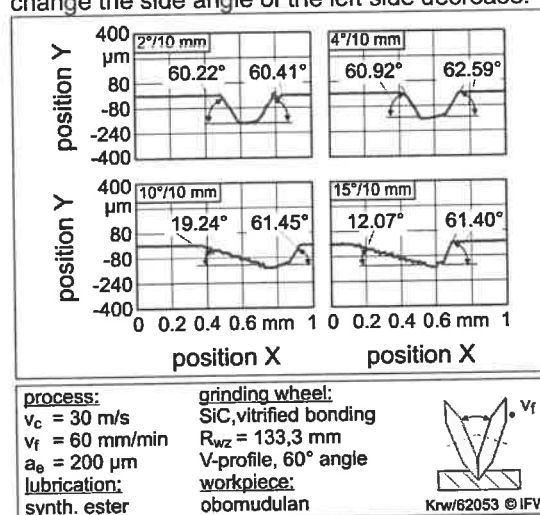


FIGURE 6. Impact of the tilt angle on the side angle

Compared to Figure 5 the influence of the tilt angle is less. For grinding of compressor blades the tilt angle is about 4° at $10\ \text{mm}$ grinding way. These basic coherences were used in order to ground a free formed surface and to structure it in a second step. The workpiece has a convex curvature in feed direction and orthogonal to the feed direction.

For structuring a SiC grinding wheel with a V profile was used. The radius of the tool path radius varied from $100\ \text{cm}$ to $80\ \text{cm}$. The ground micro profiles are demonstrated in figure 7.

The cross section shows that ground riblets depend of the position X, which is related to the tool path radius. At a position nearly $9\ \text{mm}$, or at a tool path radius of $100\ \text{cm}$, the peaks of the

geometries are trapezoid, but the geometries around 3.6 mm have a triangular form. In addition, measurements of the profile show that the height of the structures increases between 0 and 1.8 mm. The ground riblets are influenced by the kinematic although the tool path radius is higher as the calculated critical one. But the combined rotation of tilt of the grinding wheel decrease the useable tool path radius further.

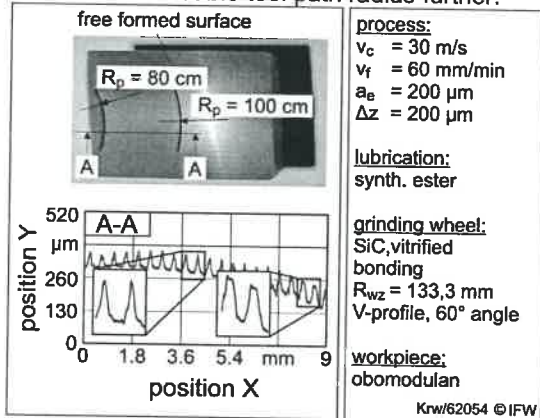


FIGURE 7. Micro pattern on a free formed surface

CONCLUSION

Riblet-structures reduce the wall near friction up to 10 %. Industrial application requires grinding of micro patterns with an aspect ratio of 0.5 on free formed surfaces in a high accuracy. The adaptation of the macro geometry of the grinding wheel to the work piece curvature increases the shape accuracy in grinding of free formed and double curved surfaces. The required micro geometry can be produced by ECDD and a shift strategy. Whereby, the grain size has an influence on the dressed profiles so that smaller grains enable sharper geometries. Furthermore, the impact of the five axes kinematic on the ground riblet-structures was investigated. If a grinding wheel moves on a circular path, the ground structures are not influenced above a critical tool path radius. This critical radius can be decreased by applying smaller grinding wheels. If the tool path radius is smaller than the critical one, the side angle of the structures reduces. As a consequence the width rises and the riblet geometry changes. Moreover, the tilt of the grinding wheel can change the side angle too, but the influence is less. The combination of rotation and tilting reduces the critical tool radius further.

Actual investigations aim for downscaling the profile geometries on metal bonded grinding

wheels and for investigating the wear behavior. Additionally, in order to react on the impact of the five axis kinematic, micro profiles are dressed, which are adapted to the movement on the grinding wheel.

Acknowledgements

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Design and Control of a MEMS Micropositioner Array for Coordinated Motion Tasks

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MEMS Array Design

In this work we present a Micro-Electro-Mechanical (MEMS) positioning array for coordinated micromanipulation tasks. The array, which can be seen in Figure 1, consists of six two degree-of-freedom (R- θ) manipulators. It is intended for use in microbiological studies, though it is versatile enough for other applications. Recent research has shown that a cell's mechanical properties may convey important information about its health. For instance, abnormalities in a fertile egg can be detected based on its elasticity [1]. Though many systems such as gripper mechanisms have been designed with cellular manipulation in mind [2], they are significantly limited in application because they are inadequate for multi-cell manipulation, and must be positioned via external means to the target cell. The new prototype array designed by the authors addresses this problem by opening up the potential for high-throughput cellular manipulation and testing. Its very large workspace means a significant number of cells can be reached without repositioning the array.

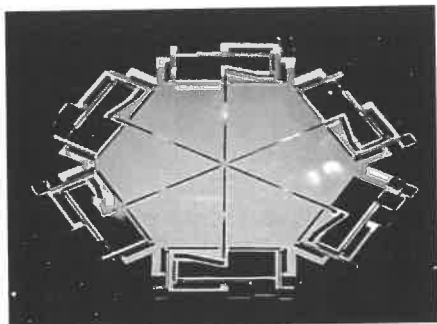


FIGURE 1. A photograph of the prototype system showing all six two degree-of-freedom manipulators in a circular arrangement

The design requirements greatly influenced the decision to use Electro-Thermal-Compliant microactuators [3]. They offer both high force and high displacement properties with a small

footprint. They are also robust to manufacturing variation and environmental contamination such as dust. The shape of the actuators in the prototype stage was modified from the standard ETC actuator topology in order to improve mechanical advantage and decrease footprint.

Figure 2 shows a drawing of a single stage with significant features labeled. Each stage's theta axis is linearly amplified by the long length of the end effector. The left and right theta actuators are antagonistically opposed. Besides allowing both directions of movement, it balances the device mechanically, and increases stiffness. Both factors decrease parasitic motion. A lever was used to amplify the displacement in the radial direction 3.6 times. Due to these amplifications techniques the stages can displace over 115 μm each axis with no sign of performance degradation.

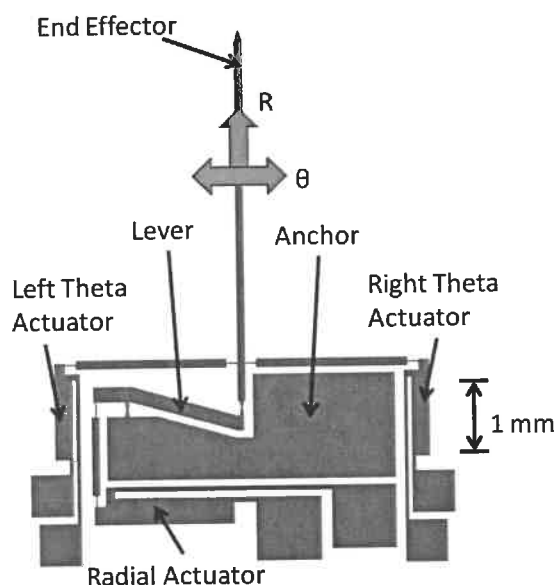


FIGURE 2. A CAD drawing of a single stage with features and axes labeled

Control Approach

Due to the difficulty associated with integrating displacement sensors into a MEMS micropositioner, a vision based feedback system was chosen. A machine vision algorithm was designed to identify the coordinates of each manipulator from a video of the array. Ordinarily the relatively slow camera frame rate would limit the speed at which control can take place. In order to address this limitation, a Kalman Filter was implemented to estimate the system in between camera frames. Each axis of each stage functions as a linear-parameter-varying first order system with a static nonlinearity relating input voltage to system gain. An iterative learning control (ILC) system was therefore used to correct for the nonlinearity as well as un-modeled disturbances. In order to demonstrate the technique of combining an estimator with an ILC system, [4] a test was carried out to track a 40 μm diameter, 3 Hz sine wave using only 10 Hz feedback. Figure 3 shows how without learning it is not possible to accurately track the reference with a linear model. The tracking after learning is shown in Figure 4. Figure 5 shows that the maximum error in both axes is less than 1 μm . Future work on improving the estimator accuracy will decrease the tracking error further.

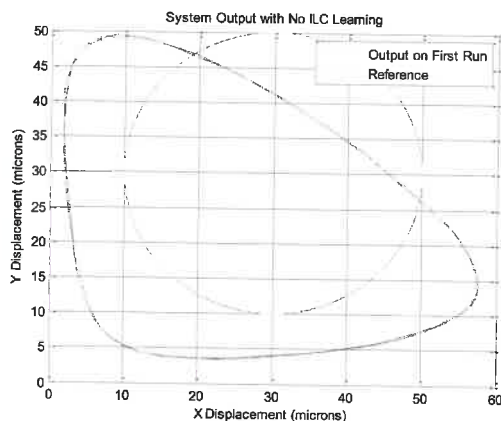


FIGURE 3. Tracking of the circle before ILC learning

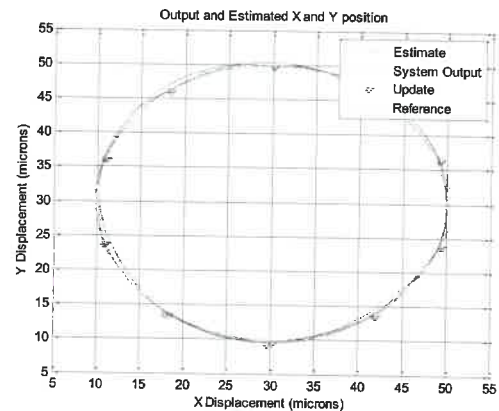


FIGURE 4. Tracking of a 3 Hz circle with 10 Hz feedback. Measurement points used to update the Kalman Filter are indicated with an *

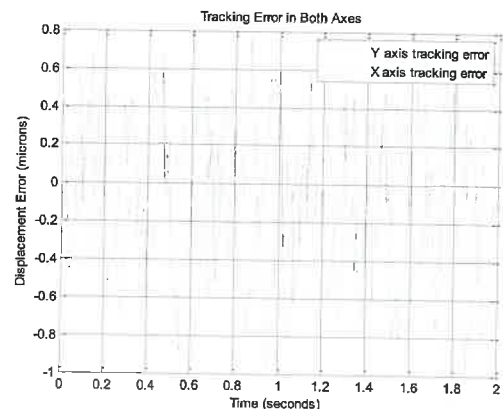


FIGURE 5. Tracking error in X and Y directions on the final ILC run

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EFFECT OF SAMPLING JITTER AND CONTROL JITTER ON POSITIONING ERROR IN MOTION CONTROL SYSTEMS

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INTRODUCTION

In a digital motion control system, there are two periodic events connecting the discrete domain of a digital controller to the continuous domain of the plant to be controlled: a feedback sampling event that samples a sensor feedback signal; and a control updating event that updates the controller's output signal via a zero-order-hold (ZOH). Generally, these two events are assumed to happen simultaneously at evenly spaced intervals of sampling period T_0 . However, in reality the sampling event intervals are not evenly spaced due to factors such as resource sharing and task scheduling. These sampling event and control event temporal deviations from the ideal timing are referred to as sampling jitter and control jitter, respectively.

Measurements have shown that jitter in motion control applications typically ranges from hundreds of nanoseconds to tens of microseconds for commercially available real-time controllers. For example, a modern digital motion controller running real-time Linux has shown several microseconds of jitter [1] and a National Instruments CompactRIO has shown 40 μ s of jitter for a 1kHz control loop [2]. The test conducted at UBC showed that the jitter on an xPCTarget controller to be 0.81 μ s RMS, and the jitter on a dSPACE DS1103 controller to be 0.16 μ s RMS. What is of interest is how much this realistic jitter affects the performance of a precision motion control system.

Simulation has been conducted to show jitter's effect on position errors [3] [4] [5]. However, there have been very few experimental results reported in literature to actually demonstrate the effect of jitter on motion system performance. One rare example is a motor speed experiment conducted by Kobayashi et al., which compared a case with fixed 0.25 ms sampling period to a case with varying sampling period from 0.25 ms to 0.375 ms [6]. Their results showed a relatively small difference in speed error for these two cases as other error sources appear to dominate the system.

Modeling work has been conducted by Boje to approximate disturbance model in the w-domain for sampling jitter and control jitter by using a Tustin approximation to convert discrete-time controllers to the w-domain [7]. Based on this approximate model he then performed simulations to show jitter caused a disturbance to act on the system.

Given the lack of analytical predictions and experimental demonstrations regarding jitter's effect on motion control system positioning error, the contributions of this paper are: (1) establishing a simplified discrete model for systems with sampling jitter and control jitter; (2) providing a formula to analytically predict jitter's effect on motion control system positioning error, without requiring simulation; (3) proposing a simple add-on jitter compensator to mitigate jitter's effect on regulation error, without requiring the existing motion controller to be changed; and (4) experimentally demonstrating the effect of sampling jitter and control jitter on positioning error for both regulation and tracking scenarios.

MODELLING of Digital Control Systems with Non-ideal Sampler and ZOH

A realistic digital motion control system is shown in FIGURE 1, which includes several timing problems introduced during real-time

implementation. First, there exists a delay τ_d between the sampling event and the controller output update event due to data-acquisition conversion times and control algorithm computation time. Second, the sampling event intervals are not evenly spaced due to factors such as resource sharing and task scheduling. These sampling event and control event temporal deviations from the ideal timing are

referred to as sampling jitter $\tau_s[k]$ and control jitter $\tau_c[k]$, respectively. Although simulations have been conducted in existing literature, there are no analytical predictions or quantitative

experimental results relating jitter to positioning error in a motion control system.

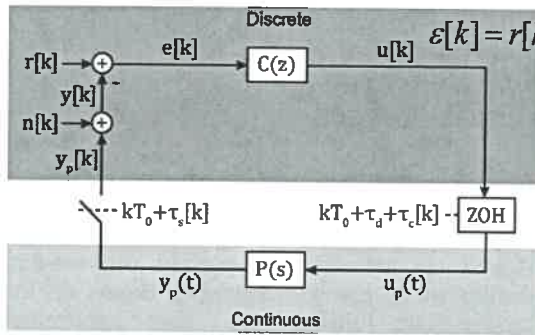


FIGURE 1: A realistic digital control feedback system with non-ideal sampler and ZOH.

To investigate the jitter effect on positioning error, this paper presents a simplified discrete model, shown in FIGURE 2, that incorporates the effect of sampling jitter and control jitter as disturbances to the control system. $\gamma[k]$ is referred to as the normalized control jitter

$$\gamma[k] = \frac{\tau_c[k]}{T_0}$$

$\lambda[k]$ is referred to as the normalized sampling jitter

$$\lambda[k] = \frac{\tau_s[k]}{T_0}$$

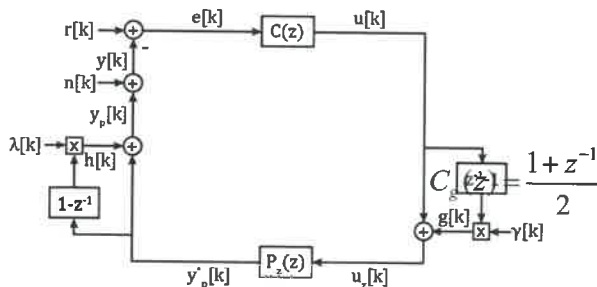


FIGURE 2: Discrete jitter disturbance model of a realistic digital control system with non-ideal sampler and ZOH.

ANALYSIS OF JITTER'S EFFECT ON POSITIONING ERROR

Based on the model in Fig. 2, analyses are carried out to determine the effect of jitter on positioning error for two scenarios: position regulation and command tracking. The positioning error (regulation or tracking) $\epsilon[k]$ is

defined as the desired plant output (reference command) minus the actual plant output sampled by an ideal sampler,

It should be noted that this definition of positioning error $\epsilon[k]$ is different from the control error $e[k] = r[k] - y[k]$ (the desired plant output minus the sampled sensor feedback), due to the presence of measurement noise and sampling jitter.

For the regulation case, the root-mean-square (RMS) of the positioning error can be calculated as

$$\sigma_\epsilon^2 = \sigma_n^2 \left(\frac{1}{2\pi} \int_{-\pi}^{\pi} \left| \frac{P_z(e^{j\Omega})C(e^{j\Omega})}{1 + P_z(e^{j\Omega})C(e^{j\Omega})} \right|^2 d\Omega \right) + \sigma_\gamma^2 \left(\frac{1}{2\pi} \int_{-\pi}^{\pi} \left| \frac{C(e^{j\Omega})(1 - e^{-j\Omega})}{1 + P_z(e^{j\Omega})C(e^{j\Omega})} \right|^2 d\Omega \right) \left(\frac{1}{2\pi} \int_{-\pi}^{\pi} \left| \frac{P_z(e^{j\Omega})}{1 + P_z(e^{j\Omega})C(e^{j\Omega})} \right|^2 d\Omega \right)$$

where σ_n^2 and σ_γ^2 are the variance of measurement noise and normalized control jitter, respectively. In this result, the first term is the regulation error contribution from measurement noise and the second term is the contribution from control jitter. The sampling jitter's effect is much less than that of the measurement noise, and thus is not included in the equation. The overall regulation error magnitude is dependent on the measurement noise, normalized control jitter, controller gain, and controller disturbance rejection. The control jitter's effect on positioning error can be attenuated by adding a jitter compensator into the motion controller

For the tracking case, the reference trajectory is assumed as a sinusoidal signal

$$r[k] = R_0 \sin(\omega_0 k T_0)$$

The control jitter induced tracking error RMS is calculated as

$$\sigma_\gamma^2 \frac{R_0^2}{2} \left| \frac{C(e^{j\omega_0 T_0})[e^{-j\omega_0 T_0} - 1]}{1 + P_z(e^{j\omega_0 T_0})C(e^{j\omega_0 T_0})} \right|^2 \left(\frac{1}{2\pi} \int_{-\pi}^{\pi} \left| \frac{P_z(e^{j\Omega})}{1 + P_z(e^{j\Omega})C(e^{j\Omega})} \right|^2 d\Omega \right)$$

The sampling jitter induced tracking error RMS is

$$\sigma_{\lambda}^2 \frac{R_0^2}{2} \left| \frac{P_z(e^{j\omega_0 T_0}) C(e^{j\omega_0 T_0}) [1 - e^{-j\omega_0 T_0}]}{1 + P_z(e^{j\omega_0 T_0}) C(e^{j\omega_0 T_0})} \right|^2 \left(\frac{1}{2\pi} \int_{-\pi}^{\pi} \left| \frac{P_z(e^{j\Omega}) C(e^{j\Omega})}{1 + P_z(e^{j\Omega}) C(e^{j\Omega})} \right|^2 d\Omega \right)$$

Therefore, both sampling jitter and control jitter can degrade position error in tracking case. Using the above formula, jitter's effect on positioning error can be predicted without simulation. A key property of these relations is that they don't require analytical models of the system (such as state space and transfer functions) and an experimentally measured plant frequency response is sufficient to calculate the effect of jitter.

EXPERIMENTAL RESULTS

Experiments are performed to validate the model and analysis of the jitter disturbance effect presented in this paper. They are conducted on an improved version of the fast-tool servo (FTS) presented in [8]. This FTS can achieve 50 μm stroke, 1.4 nm positioning error, and 750 g acceleration in continuous operation. The digital control hardware is a custom real-time computer made of high-performance digital signal processors and a field programmable gate array [9]. This computer's jitter is less than 6 ns RMS. This performance is significantly better than commercial controller hardware, which typically is limited to <100 kHz sampling rate and >100 ns jitter. Various amount of jitter is added into feedback sampling and control output ZOH to investigate their effect on positioning errors.

In the case of position regulation, sampling jitter has a negligible effect on positioning error, as shown by FIGURE 3. In comparison, FIGURE 4 shows that control jitter can significantly degrade regulation performance, with regulation error increasing from 4.0 nm RMS at zero control jitter to 7.7 nm RMS at 8% control jitter. Adding the presented jitter compensator to the existing motion controller reduces the regulation error back to 4.7 nm RMS, despite the 8% control jitter.

In position tracking experiment, the reference command is a 6 kHz sinusoidal signal with a 4 μm peak-to-valley amplitude: $r[k] = 2 \sin(2\pi \times 6000 \times kT_0) \mu\text{m}$. The

sampling period T_0 is 4 μs . Generally, tracking at such a high frequency will result in the

dominant tracking error contribution coming from the reference command component. In order to show the tracking error contributed by jitter, the trajectory induced tracking error should be eliminated by increasing the controller gain at 6 kHz to infinity. This can be done by adding an adaptive feed-forward cancellation (AFC) controller. Fig. 5 shows the tracking experiment

results for no jitter, with and without $C_{AFC}(z)$. After implementing the AFC controller, the tracking error was reduced by a factor of nearly 1000, from 1.4 μm RMS to 1.6 nm RMS, which is close to the measurement noise floor.

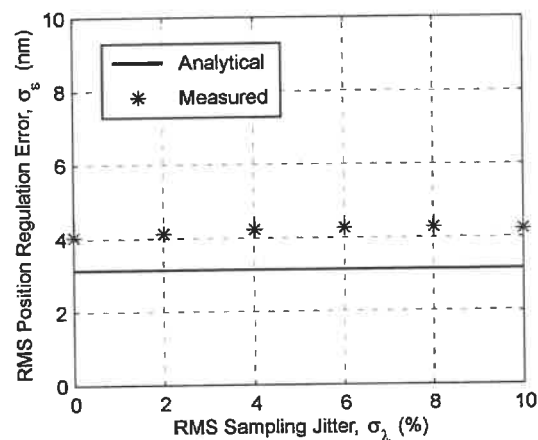


FIGURE 3: Measured and analytical RMS regulation error comparison for various amounts of sampling jitter.

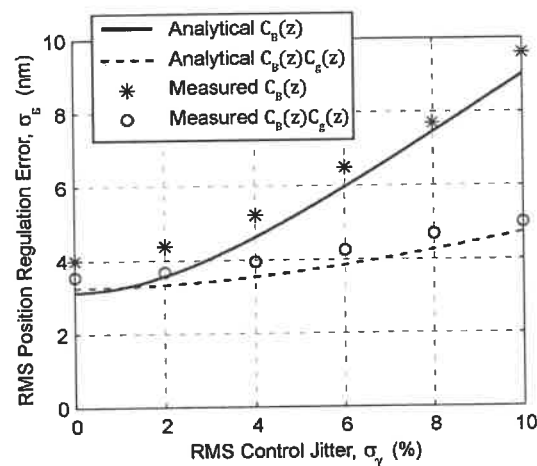


FIGURE 4: Measured and analytical RMS regulation error comparison for various amount of control jitter, with and without the jitter compensator.

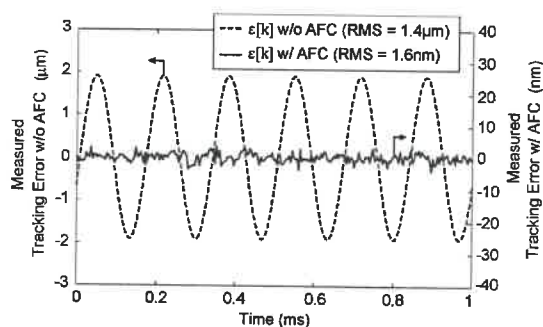


FIGURE 5: Tracking error experimental results for no jitter with and without AFC.

The tracking error results with various amounts of sampling jitter are shown in Fig. 6, which compares the analytical RMS tracking error to the experimentally measured RMS tracking error for sampling jitter ranging from 0% to 10%. Notice that as sampling jitter approaches zero, the measured error approaches 1.6nm RMS, nearly the measurement noise floor for the FTS system. In all cases, the analytical results predict the trend of the experimental results with a small amount of mismatch. This is believed to be related to the FTS plant non-linearity which was not modeled and included in the analysis. These experimental results also indicate that the sampling jitter disturbance can become the dominant source of positioning error, particularly for high-bandwidth precision motion control systems.

Fig. 7 shows the tracking error results for control jitter, comparing the analytically predicted RMS tracking error to the experimentally measured RMS tracking error for control jitter ranging from 0% to 10%. In all cases the results match very well, validating the presented models and analytical results for control jitter. In addition, the experimental results also demonstrate that control jitter can significantly degrade positioning performance.

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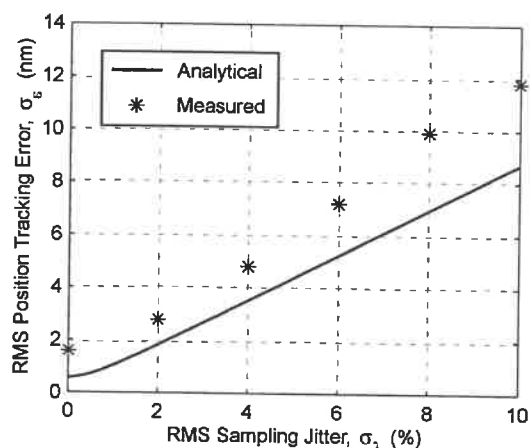


FIGURE 6: Measured and predicted tracking error comparison for various amounts of sampling jitter

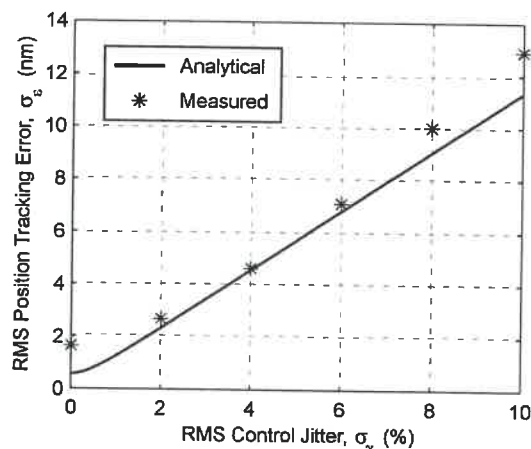


FIGURE 7: Measured and predicted tracking error comparison for various amounts of control jitter.

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MODEL-BASED AUTOMATED CONTROLLER TUNING

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INTRODUCTION

Though more than half a century has passed since the first tuning rules for proportional integral derivative (PID) controllers were defined, the PID controller is still omnipresent in industry [1] and a myriad tuning algorithms for PID laws have been developed. Several methods and architectures have been explored for these controllers as summarized in a handbook [2]. However, controller tuning remains primarily a heuristic art and requires substantial engineer time.

Auto-tuning laws for PID controllers have attracted researchers for a long time. With the introduction of optimization-based tuning algorithms, *Automated controller tuning* found renewed interest, such as *Iterative Feedback Tuning (IFT)* [3] and mixed virtual reference feedback tuning - iterative feedback tuning (VRFT-IFT) [4]. Since these controller tuning algorithms are purely experimental and use no model information, they provide no analytical guarantees of performance and furthermore require an operator to be present to avoid unforeseen events as well as to analyze the experimental data. On the other hand, model-based control design algorithms, though powerful for analytical results, often suffer from conservativeness since they assume large uncertainty sets and worst case performance. This leads us to believe that paradigms for optimal fusion of information from system models and experimental data will finally lead to the design of *automated model-based controller tuning algorithms*.

The critical challenge in designing an automated tuning algorithm is to guarantee that the system remains stable for all possible choices of the tuned parameters. Typically, one may decide to check for stability at each step of the tuning process. Alternatively, by choosing a sufficiently small 'step size' for parameter update, this problem can be addressed. However, this makes the tuning process longer and complex. Therefore, we use the Youla parameterization, a way to parameterize all controllers that stabilize the system [5], to guarantee system stability for a linear time

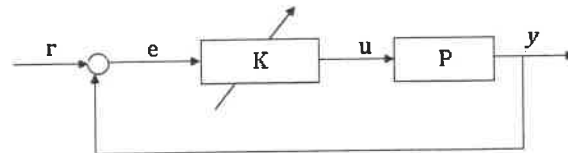


FIGURE 1. Typical unity feedback control loop

invariant(LTI) system when the controller is being tuned. As a consequence, the closed loop system always stays stable if the controller parameters are tuned within the frame of Youla parameterization.

In this paper, we propose a model-based auto-tuning (MBT) method. As in IFT, the goal of the MBT method is to minimize a given cost function (say, the L_2 norm of the error and the applied control effort) through the tuning of a set of controller parameters. The *controller parametrization*, however, is based on the model of the plant, while the tuning algorithms use *experimental data* to obtain the search direction to minimize the cost function. In this way, we marry the concept of experimental tuning with model-based controller design. This eliminates the issue of checking for stability at each step. Eventually, these automated controller tuning algorithms will realize the goal of "plug and play" precision motion control systems.

PROBLEM FORMULATION

The traditional block diagram for a feedback system is shown in Fig. 1, where K is the controller and P is the plant. r , e , u and y are the reference, the tracking error, the control input and the output signals, respectively.

The typical controller tuning problem can be stated as "*iteratively determine $K(s)$ (causal) such that the cost function $J(e, u)$ is minimized while guaranteeing that the feedback system in Fig. 1 is stable; i.e. $(I - PK)^{-1} \in RH_\infty$* ". – (P1)

YOULA PARAMETER-BASED TUNING STRATEGY

In this section, we will reform problem (P1) into a Youla parameterization framework and tune pa-

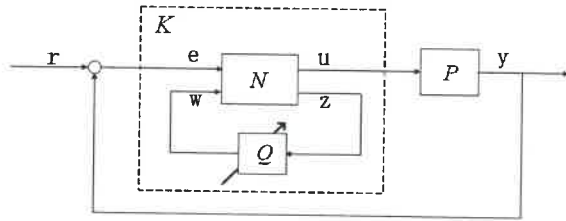


FIGURE 2. Structure of stabilizing controllers in unity feedback control loop

parameters in this framework to minimize a cost function.

Although the closed-loop feedback system in Fig. 1 is straightforward, it can not guarantee the stability of the system while parameters of the controller K are being tuned. We use Youla parameterization to avoid the problem of stability in auto-tuning. The Youla parameterization is an approach to parametrize *all internally stabilizing feedback controllers* for a given stable or unstable plant [5], which ensures that the closed loop system remains stable while it is being tuned.

To present the Youla parameterization, the feedback system in Fig. 1 is first reformulated to Fig. 2. The controller K in dash line box is equivalent to the one in Fig. 1, but it has been split into two parts: a *fixed* observer-based estimator N and a tunable Youla parameter, Q . It is crucial to note that *all* stabilizing controllers can be written in this form. The benefit of Youla parameterization is that the closed-loop stability depends only on the stability of Youla parameter, Q , which can be manipulated easily in comparison with the stability of the whole loop in Fig. 1.

Thus, the typical controller tuning problem (P1) can be restated as “*iteratively determine $Q(s)$ (stable) such that the cost function $J(e, u)$ is minimized, i.e. $Q \in RH_\infty$ ”.* – (P2)

The block diagram in Fig. 2 is wrapped to form in a linear fractional transformation (LFT) framework, as shown in Fig. 3, where G is a generalized plant and K is the controller.

The generalized plant G is derived from the original plant P , as shown in Eq. 1. It has two inputs (the reference input r and the control effort u) and two outputs (the plant output y and error e).

$$G \triangleq \begin{bmatrix} 0 & P(s) \\ 1 & -P(s) \end{bmatrix} \equiv \left[\begin{array}{c|cc} A & B_1 & B_2 \\ \hline C_1 & D_{11} & D_{12} \\ C_2 & D_{21} & D_{22} \end{array} \right] \quad (1)$$

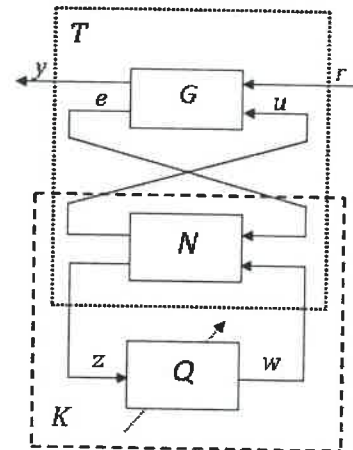


FIGURE 3. Structure of stabilizing controllers [6]

The observer-based estimator N is expressed in Eq. 2, in which L and F are arbitrarily chosen as long as $(A + LC)$ and $(A + BF)$ are Hurwitz.

$$N \triangleq \begin{bmatrix} N_{11}(s) & N_{12}(s) \\ N_{21}(s) & N_{22}(s) \end{bmatrix} \quad (2)$$

$$\equiv \left[\begin{array}{cc|cc} A + LC_2 + B_2F & -B_2F & -L & B_2 \\ LC_2 & A & -L & B_2 \\ \hline F & -F & 0 & I \\ -C_2 & C_2 & I & 0 \end{array} \right]$$

As indicated in Fig. 3, the system can be treated as a combination of a generalized plant G and a controller K , which is matched with the blocks of the canonical unity feedback loop in Fig. 2 correspondingly. Meanwhile, it can also be viewed as the block T and the tunable Youla parameter Q . The block T (consisting of P and the fixed part N) has inputs r and w and outputs y and z . The derivation of Eq. 2 and Eq. 3 can be found in [6].

$$T \triangleq \begin{bmatrix} T_{11}(s) & T_{12}(s) \\ T_{21}(s) & T_{22}(s) \end{bmatrix} \quad (3)$$

$$\equiv \left[\begin{array}{cc|cc} A + B_2F & -B_2F & B_1 & B_2 \\ 0 & A + LC_2 & B_1 + LD_{21} & 0 \\ \hline C_1 + D_{12}F & -D_{12}F & D_{11} & D_{12} \\ 0 & C_2 & D_{21} & 0 \end{array} \right]$$

the error e is given by:

$$E(s) = [T_{11}(s) + T_{12}(s)Q(s)T_{21}(s)] R(s) \quad (4)$$

There are two advantages of dividing the system into block T and Q : (1) Block T is an unchanging part, while Q is a tunable part and Q 's stability determines the closed loop stability. (2) The error e is an affine function of Q with respect to T , which can be seen from Eq. 4. It is those two advantages that make the auto-tuning possible.

The Tunable Youla Parameter 'Q'

As stated, any stable Q parameter will not violate the stability of the system. Therefore, there is an infinite dimensional search space for the optimal controller. To reduce the search space, we may choose a parameterization of the form

$$Q(s) = \sum_{i=1}^j k_i q_i(s) \quad (5)$$

where q_i are all fixed stable transfer functions, the tunable parameters $k_i \in \mathbb{R}$ and $j \in \mathbb{N}$. $q_i(s)$ thus form the basis of our search space. Candidate basis transfer functions $q_i(s)$ can be obtained by back-calculation from standard tuned control laws, e.g., Ziegler-Nichols (ZN) tuning rule, Internal Model Control (IMC) method, LQR, etc.

Since our candidates $q_i(s)$ are back-calculated from some typically tuned control law $K_i(s)$, the conversion from K to Q is significant. From the equivalence of the controller K and the combination of N and Q shown in Fig.2 and the expression of N in Eq. 2, K can be related to N and Q :

$$K(s) = N_{11}(s) + N_{12}(s)Q(s)[1 - Q(s)N_{22}(s)]^{-1}N_{21}(s) \quad (6)$$

Eq. 6 is then regrouped to get Q 's expression:

$$Q(s) = [K(s) - N_{11}(s)][(K(s) - N_{11}(s))N_{22}(s) + N_{12}(s)N_{21}(s)]^{-1} \quad (7)$$

Cost Function

Before defining a cost function, we first introduce e , an M -sample sequence of the error $e(t)$ over an entire experimental run (M is the number of samples in a single run). To be specific, $e = [e(0) \ e(1) \ e(2) \ \dots \ e(M-1)]^T$. Vector r and u are defined in the same way.

Cost functions using weighted tracking error e and control effort u can be considered, such as $J = \frac{1}{2}e^T W_e e + \frac{1}{2}u^T W_u u$, where $W_e, W_u > 0$. For illustration, we choose the cost function J to be square of the L^2 norm of error e over a single run: $J = \frac{1}{2}e^T e$.

Gradient Search Method

The gradient search method is an approach to seek the extremum of a function. We use it to look for a local minimum of cost function J . Because the tuning of parameters Q is within the frame of Youla parameterization, we are able to use gradient search method to tune the controller automatically without worrying about the closed-loop system's stability. The parameters k_i are updated through Eq. 8.

$$k_i^{(n+1)} = k_i^{(n)} - \epsilon(\nabla_{k_i} J)^{(n)} \quad (8)$$

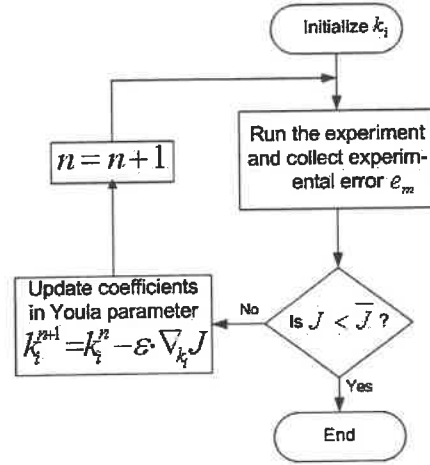


FIGURE 4. Flow chart of MBT algorithm

where ϵ is the step size and can be either fixed or variable. $\nabla_{k_i} J$, the gradient of J w.r.t. k_i , is derived from the expression of the error in Eq. 4, and the parameterized search space:

$$\nabla_{k_i} J = (T_{12} q_i T_{21} r)^T e \quad (9)$$

where T_{12} is Toeplitz matrix form of the linear, time-invariant and causal system described by transfer function $T_{12}(s)$ from Eq. 3.

$$T_{12} = \begin{pmatrix} t_{12}(0) & 0 & \dots & 0 \\ t_{12}(1) & t_{12}(0) & & \vdots \\ \vdots & & \ddots & 0 \\ t_{12}(M-1) & t_{12}(M-2) & \dots & t_{12}(0) \end{pmatrix} \quad (10)$$

where $t_{12}(m), m \in [0, M-1]$ is the impulse response of $T_{12}(s)$. The matrices T_{21} and q_i are obtained in the same way from $T_{21}(s)$ and $q_i(s)$. The error e in Eq. (9) is then replaced by measurement from the plant e_m , which is collected from an experimental run of the closed-loop system using the parameters k_i^n in last iteration.

Remark: For a faster and more efficient search, a better search method can be applied, like Newton's Method. Constraints on inputs, outputs etc. may also be handled by interior-point methods.

ALGORITHM

The flow chart in Fig. 4 shows how our MBT method works. We start with an initial controller parameter set, then run the experiment to collect the error. The cost function is calculated based on the error, if the cost function J is still needed to be optimized, a further k_i is obtained using gradient search method. The process is repeated till a satisfactory cost function $\bar{J}$ is achieved.

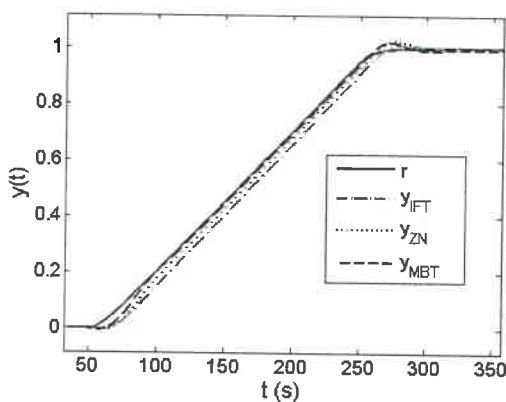


FIGURE 5. Plant output for a raster scan reference using a ZN-tuned controller, an IFT-tuned controller and the MBT tuned controller.

APPLICATION

The MBT method is tested on a simulated system from Lequin et al.[3], which is a non-minimum phase plant $P(s) = \frac{-5s+1}{(10s+1)(20s+1)}$.

The Youla parameter Q here is a linear combination of three stable controller candidates $q_i(s)$ as: $Q(s) = k_1 q_1(s) + k_2 q_2(s) + k_3 q_3(s)$, where k_i ($i = 1, 2, 3$) are the parameters to be tuned, q_1 is a proportional gain, q_2 is back calculated from a controller tuned with ZN method, and q_3 is back calculated from a controller designed with IMC method. For illustration in the extended abstract, we use a raster scan reference.

Fig. 5 shows the output of the system using the MBT tuned controller, an IFT tuned controller, and a ZN-tuned controller for a raster scan reference. The MBT tuned control law outperforms the IFT and ZN tuned controllers. A plot of the cost function decrease over tuning steps is shown in Fig. 6. The example illustrates that through the model-based tuning process we can exceed the performance obtained from either model-based or purely experimental tuning.

CONCLUSION

Automated controller tuning has the potential to realize the ideal of *plug and play* hardware for precision motion control systems. Algorithms that *intelligently* combine experimental data with model-based design are a critical component of these automated controller tuning systems. A *necessary* feature of any tuning algorithm is the guarantee that the system remains stable for any acceptable choice of the tuning parameter. In an

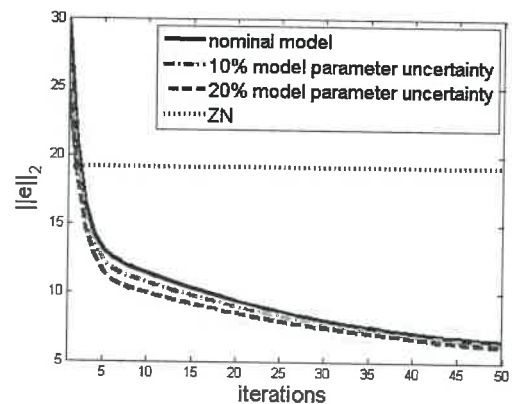


FIGURE 6. Cost function (2-norm of error) plotted against tuning steps. The cost for the ZN-tuned controller is shown as a benchmark.

operator-less environment, this is difficult to guarantee. However, by using the model information through the Youla parameterization, we can restrict our search space to only stabilizing controllers. Based on this idea, a model-based tuning technique was developed in this paper, which utilizes experimental data to tune the Youla parameter in order to minimize a prescribed cost function.

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PRECISION MOTION CONTROL OF ULTRAHIGH-ACCELERATION AND HIGH-VELOCITY LINEAR DRIVE MECHANISM

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INTRODUCTION

In precision industrial machines, for example, machine tools and measuring machines, their motion characteristics greatly influence the machine performances. High motion accuracy and high speed characteristics are important motion characteristics for the machines. These characteristics essentially depend on those of linear drive mechanisms in the machines and their improvements are strongly required. Particularly, the required speed characteristics are becoming strict increasingly [1]. The speed characteristics are determined by the hardware performance of the linear drive mechanism. In recent years, linear motors are widely used for high speed precision machines. Because they are direct-drive and have the advantages of small driven inertia, no speed limitation by the mechanical structure, ease to control. Although some high speed precision industrial machines have the combinations of rotary motors and lead screws, their acceleration is often lower than 19.8m/s^2 ($\approx 2\text{G}$) and the velocity is lower than 2m/s . For higher acceleration and higher velocity motion, linear motors are used [2-4]. In the market, there are linear motors of which maximum acceleration is about 40G [5][6]. The purpose of this research is to realize an ultrahigh speed linear drive mechanism with a long working range for precision industrial machines. For this purpose, the ultrahigh thrust density electromagnetic linear motor has been designed and fabricated. A prototype linear drive mechanism with the linear motor has shown the acceleration higher than 100G and the velocity higher than 11m/s . This paper describes precision motion control of the ultrahigh-acceleration and high-velocity linear drive mechanism.

ULTRAHIGH-ACCELERATION AND HIGH-VELOCITY LINEAR DRIVE MECHANISM

Construction of the Prototype

Figure 1 shows a schematic view of a linear drive mechanism designed for high thrust

density. Figures 1(a) and 1(b) are the view from traveling direction and the side view, respectively. As an actuator, a moving permanent magnet linear synchronous motor (MPM LSM) is used. The linear motor has a double-sided structure and cored electromagnets (EMs) for large thrust. The moving PM type is suitable for high speed and easy to control. The mover of the MPM LSM comprises eighteen PMs, spacers and a frame for holding the PMs separated by the spacers. The shapes of the PMs and the EMs were optimized for high thrust density using a magnetic field analysis program. The pitch of PMs in the mover has been adjusted so that the thrust ripple has become small.

Figure 2 shows a prototype of the linear drive mechanism. The mover is supported by linear ball guides. The length and weight of the mover are 440mm , 4.68kg respectively. The length of stator is 2.03m . The working range is about 1.5m . The average thrust of the linear motor is $3,463\text{N}$ when the current signals of which

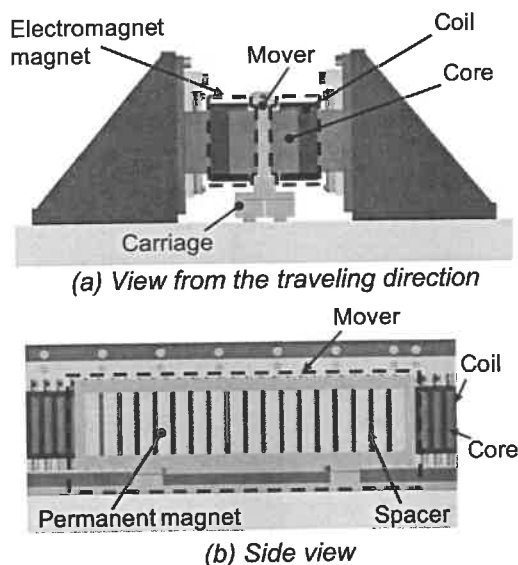


Fig.1 Schematic view of the experimental MPM LSM

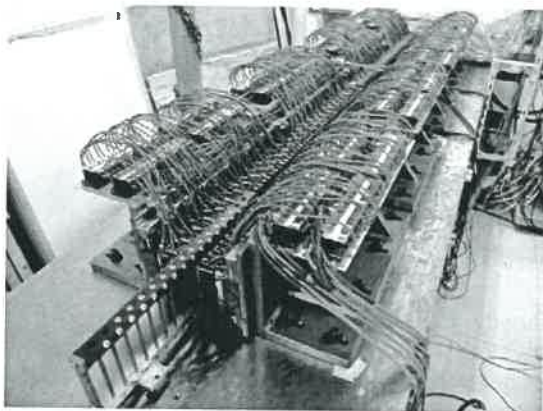


Fig.2 Prototype of ultrahigh-acceleration and high-velocity linear drive mechanism

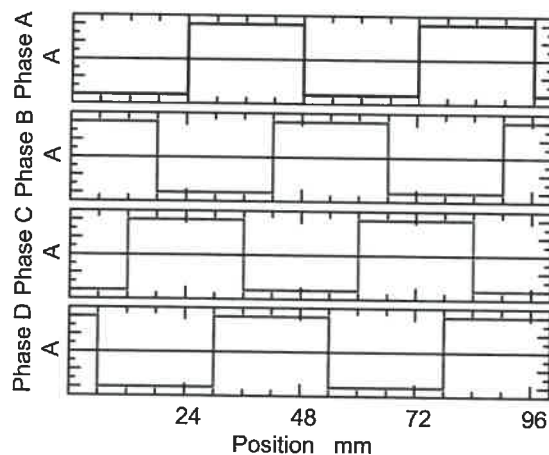


Fig.3 Driving current command signals for the prototype

amplitudes are 20A are applied to each coil in the EMs. The displacement of the mover is measured with a laser position transducer (ZLM series from JENAer Meßtechnik GmbH. Resolution: 79.1 nm).

Driving Characteristic

The prototype MPM LSM is driven by four current commands which have rectangular waveform shown in Fig.3. They have the same amplitude, but their phases differ each other. The phases depend on the position of the mover. The dynamic response of the prototype was measured using rectangular wave current commands of which amplitude was 20A. The measured response is shown in Fig.4. The experimental results indicate that the prototype can move at the acceleration higher than 779m/s^2 ($=79.4\text{G}$) and at the velocity higher than

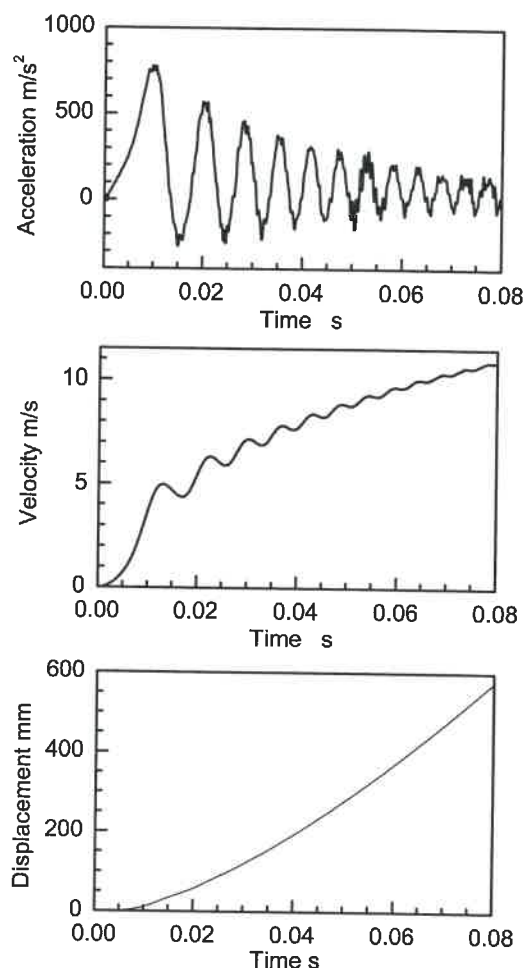


Fig.4 Open-loop driving response

11.1m/s. The acceleration and velocity are much higher than those of conventional high speed linear drive mechanisms. The acceleration is vibrated and the amplitude of the vibration is attenuated by the limitation of power supply performance, that is, the applied voltage limitation and the large back EMF.

CONTROL SYSTEM DESIGN

The direct drive feature of linear motors provides advantages of high speed and high thrust density. However it makes the motion characteristics of linear motors sensitive to friction, thrust gain, and cogging force. In the controller design, these were taken into account.

(i) Friction effect

Friction is a representative factor which deteriorates motion control accuracy. In order to compensate the bad friction effect, model-based friction estimators are often used as feedforward

or feedback elements. There are many friction models used in the elements. The feedforward element used in this paper is expressed as a function of the mover velocity and determined from the measured responses of the mover. In general, feedforward elements are effective for high speed motion. However they are sensitive to the characteristic change and friction characteristics tend to change. For increasing the robustness to the change, the disturbance observer is also added in the control system.

(ii) Nonlinearity of the thrust gain

The thrust gain of linear motors with cored EMs depends on the applied current. In the designed controller, the linearization element is used to cancel the influence of the applied current on the thrust gain. The thrust ripple also causes the variation of the thrust gain. In the prototype, it makes the variation of the thrust smaller than 8%. In this paper, its compensation is one of the aims of the disturbance observer.

(iii) Cogging force

Cogging force is also a representative factor to be compensated for accurate motion in linear motors with cored EMs. The cogging force can be easily estimated by the disturbance observer and expressed as a function of the position. In this paper, the feedforward element based on the estimated cogging force is used to eliminate the cogging force effect.

Figure 5 shows the block diagram of control system for the linear drive mechanism. The controller comprises a two-degree-of-freedom (2DOF) compensator which is often used as a high response controller and additional elements. In the 2DOF compensator, a PID element works as a basic compensator and a FF element is used for reducing the bad effect of dynamic characteristics of the mechanism. The FF element includes the friction model. The validity of the friction model in microscopic range greatly influences the motion accuracy.

Figure 6 shows two friction models. Figures 6(a) and 6(b) represent a standard model of Coulomb friction and a proposed friction model determined from measured microscopic characteristics, respectively. In the next section, the control performances with the FF elements based on the friction models in Fig.6 are compared. The disturbance observer also works for compensation of the friction effect. The linearization compensator is added to cancel the nonlinearity in the thrust gain.

For compensating the cogging force, the cogging force compensator is added. The

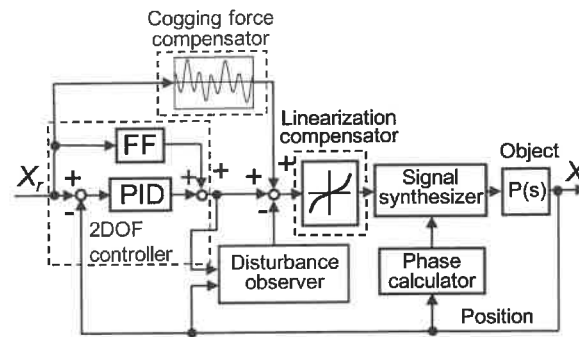


Fig.5 Control system

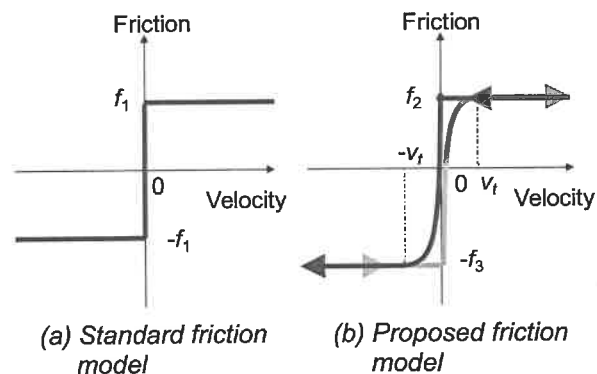


Fig.6 Friction model

disturbance observer is also used for it. In motion control, the disturbance observer is often used for reducing unknown characteristics including modeling errors. The disturbance observer is effective to cancel the bad influence which changes slowly.

EXPERIMENTAL RESULTS

Figure 7 shows the comparison results of the control system responses with the different friction models. The input signal to the control system was a sinusoidal input of which the amplitude and the frequency are 10mm and 0.5Hz, respectively. The controllers used include a PID compensator which has fixed gains and the cogging force compensator based on the periodicity of the cogging force.

The significant stick motion error is caused by the control system with the basic friction model, but not by the control system with the proposed friction model. This indicates that the proposed friction model is useful to eliminate the stick motion.

In order to reduce the tracking error, a gain scheduled PID compensator and an improved cogging force compensator were implemented

to the controller. Figure 8 shows the measured response of the improved controller to a sinusoidal input and the tracking error with the input. The input amplitude and the frequency are 100mm and 0.5Hz, respectively. The tracking error is smaller than 1 μ m.

CONCLUSION

In this paper, the motion control results of the ultrahigh-acceleration and high-velocity linear drive mechanism were introduced. The linear drive mechanism has the ability to move at the acceleration higher than 79.4G and at the velocity higher than 11m/s. For the mechanism, the controller comprising the 2DOF compensator, the disturbance observer and the cogging force compensator was designed. The experimental result shows that the controller are effective to reduce the tracking errors and that the controller makes the tracking error smaller than 1 μ m.

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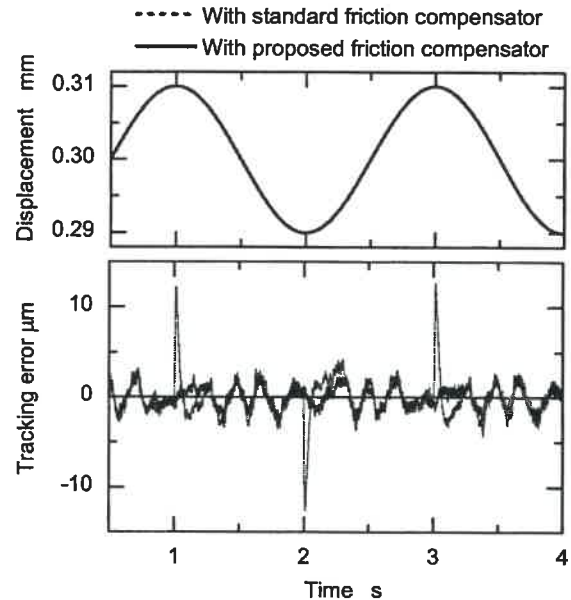


Fig.7 Tracking responses to a sinusoidal input (amplitude: 10mm, frequency: 0.5Hz)

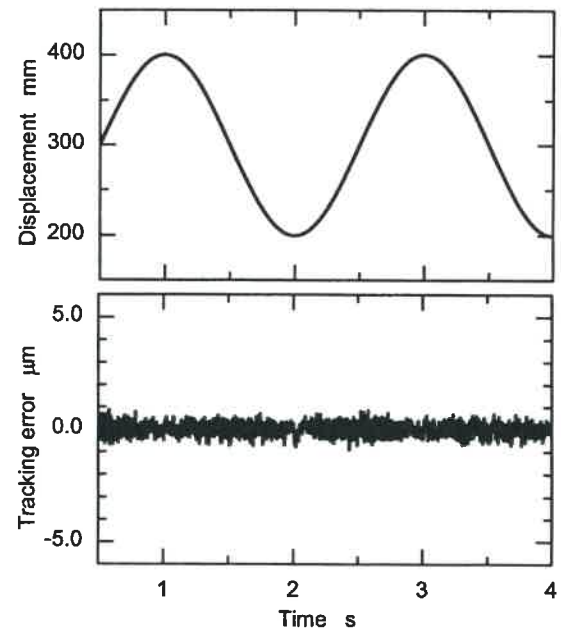


Fig.8 Tracking response to a sinusoidal input (amplitude: 100mm, frequency: 0.5Hz)

Harmonic disturbance rejection in tracking control of ball screw drives

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INTRODUCTION

Despite the recent advances in the design and manufacturing of linear motors, ball screw drives are still used extensively by the machine tool industry. Compared to linear motors, ball screws cost less and their inherent gearing ratio makes their closed-loop control more robust against load variations and cutting force disturbances. However, the motion delivery in ball screw drives is based on mechanical contact and imperfections in the mating components directly result in the deterioration of the achievable dynamic accuracy. For example, errors in the pitch angle (i.e. position dependent lead variations) cause harmonic positioning errors, which can occur at high frequencies when the drive is in high speed motion. Misalignments and grating errors in the encoders may also have a similar effect, which could be difficult to correct using classical feedback techniques, especially when the frequency of the error harmonics is beyond that of the loop bandwidth. Although model-based feedforward approaches may be used for correction purposes [1], when there are variations in the error characteristics, e.g. a change in the lead error profile due to thermal deformations or wear on the ball screw, then such compensation techniques may become ineffective.

On the other hand, machining forces in operations like milling have a periodic nature, where the first few harmonics are typically beyond the closed-loop bandwidth. In this case, the best that a typical servo controller can do is to counteract the average value of the cutting force, while leaving the high frequency components unaccounted to cause vibrations and positioning errors. The ability to counteract these harmonics would undoubtedly improve the dynamic positioning accuracy of the drive system, and therefore the quality of parts produced on the machine tool.

This paper presents an adaptive harmonic cancellation technique for ball screw drives,

which successfully mitigates the effects of lead and encoder errors, and also high frequency cutting forces, thereby resulting in more accurate motion during high travel speeds and machining. Following a brief literature review, the design methodology and experimental results are discussed.

LITERATURE REVIEW

Harmonic disturbance cancellation is a problem encountered in many engineering fields, and there have been a number of solutions proposed in literature. Among them are internal model control [2], repetitive control [3], [4], and Adaptive Feedforward Cancellation (AFC) [5]. In internal model control and repetitive control, the controller gain is infinite at the frequency of disturbance because the controller transfer function has a pair of poles on the imaginary axis corresponding to that frequency. Therefore, in these methods, it is assumed that the fundamental disturbance frequency is known and not varying. Then, the sampling frequency needs to be adjusted so that it is an integer multiple of the disturbance frequency [4]. In AFC, on the other hand, the correction signal is not generated within the control feedback loop. Instead, an external block is used to adaptively estimate the magnitude and phase of the "equivalent" disturbance causing the harmonic error, and inject an appropriate cancellation signal into the control loop to cancel its effect out. Recently [6], it has been shown that when the AFC scheme is applied to reject disturbances of time-varying frequency, it is equivalent to a linear time-varying compensator. Based on this principle, a design technique similar to the one in [7] has been extended in this paper for improving the dynamic accuracy of ball screw drives.

BALL SCREW DRIVE TEST-BED

The ball screw setup is shown in Fig.1. The first axial mode of the setup is at 133 Hz, and the first torsional mode is at 420 Hz. The ball screw diameter and pitch are both 20 mm. Feedback is

provided both by a rotary encoder mounted on the ball screw close to the motor, and by a linear encoder installed on the table. The ball screw is actuated by a 3 kW AC servomotor through a diaphragm type coupling.

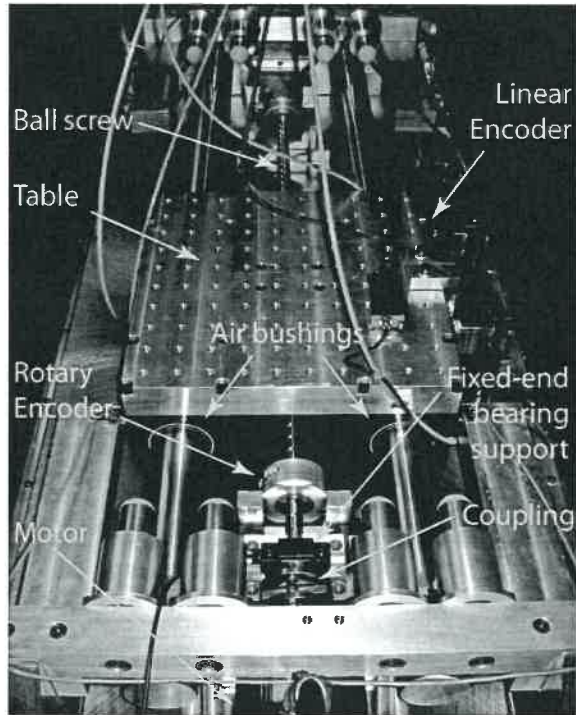


FIGURE 1. Ball screw drive test-bed.

DESIGN METHODOLOGY

Fig. 2 shows the overall control scheme. The main feedback loop is closed using pole-placement control [8], which achieves a wide cross-over frequency (~ 100 Hz) by applying active vibration damping. This loop has a phase margin of $PM \cong 30^\circ$ and is considered as a new plant upon which the AFC controller is designed. Since successful AFC requires the closed-loop system to have near-perfect tracking characteristics, the command following bandwidth has been widened by a Zero-Phase Error Tracking Controller (ZPETC) [9], which realizes stable inversion of the open-loop model, and a model based tracking error prediction and compensation scheme, which further mitigates the following errors. The model-based tracking error compensator consists of a duplicate of the pole-placement controller acting upon a two-mass model of the ball screw drive. The two-mass model parameters used in the design of pole-placement controller, and tracking error

compensator were identified using the method described in [10] and [11].

To show how the compensation signal generated by this estimator ($\hat{e}$) improves the final tracking accuracy, we first show how the injection of this signal influences the tracking error. If the inner plant has a tracking transfer function G_{track} , then the tracking error of the uncompensated system will be:

$$e = (1 - G_{track})x_r = G_e x_r \quad (1)$$

After injecting the compensation signal, the tracking error will be changed as follows:

$$\begin{aligned} (e)_{\text{modified}} &= x_r - G_{track}(x_r)_{\text{modified}} \\ &= x_r - G_{track}(x_r + \hat{e}) \end{aligned} \quad (2)$$

Assuming $\hat{e} \approx e$:

$$\begin{aligned} (e)_{\text{modified}} &= x_r - G_{track}(x_r + (1 - G_{track})x_r) \\ &= (1 - G_{track} - G_{track}(1 - G_{track}))x_r \\ &= (1 - 2G_{track} + G_{track}^2)x_r \\ &= (1 - G_{track})^2 x_r = (G_e)^2 x_r = (G_e)_{\text{modified}} x_r \end{aligned} \quad (3)$$

And because $G_e \ll 1$:

$$(G_e)_{\text{modified}} < G_e \quad (4)$$

Therefore, adding the estimated tracking error to the reference trajectory attenuates the error transfer function.

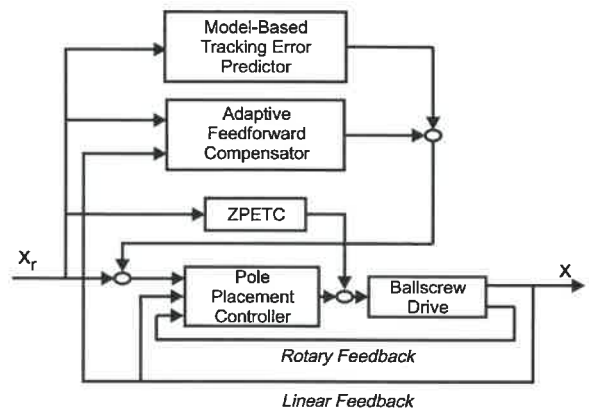


FIGURE 2. Overall control scheme including PPC as main controller and AFC as a harmonic disturbance compensator.

Although more elaborate inversion methods may be used when the full command history is available, such as applying forward and backward Fourier transforms, in a practical machine tool implementation the controller can only look ahead a few samples at a time to determine the best possible input sequence.

In the AFC design, the tracking error $e(t)$ is first amplified by a gain g_n and then de-modulated by harmonic signals generated at the disturbance frequency ω_n with phase advance parameter ϕ_n as shown in Fig. 3:

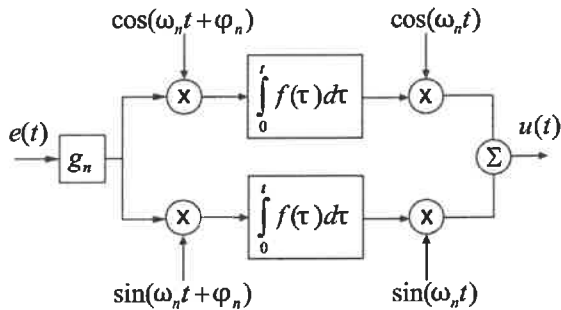


FIGURE 3. AFC scheme. [7]

A parameter adaptation law is then applied to calculate the equivalent cancellation signal:

$$u(t) = a(t) \cdot \sin(\omega_n t) + b(t) \cdot \cos(\omega_n t) \quad (5)$$

The transfer function of the linear time-invariant system equivalent to the adaptive scheme represented in Fig. 3 is:

$$\frac{U(s)}{E(s)} = g_n \frac{s \cos(\phi_n) + \omega_n \sin(\phi_n)}{s^2 + \omega_n^2} \quad (6)$$

Similar to tuning rules detailed in [7], here the phase advance parameter was set equal to the phase of the inner closed-loop system at the frequency of disturbance. In addition, actuator saturation also needed to be avoided while selecting the adaptation gains. This required additional simulations to be conducted during the design phase. Currently, analytical methods are being investigated for tuning the AFC in the presence of saturation limits.

EXPERIMENTAL RESULTS

The capability of the developed controller in rejecting periodic disturbances due to lead and

encoder errors was tested by conducting high-speed tracking experiments at 1 m/s.

In these experiments, the de-modulator signals were generated using the feedback from the rotary encoder. In other words, $\sin(\omega_n t + \phi)$ and $\cos(\omega_n t + \phi)$ terms in Fig. 3 were replaced by $\sin(\Theta(t) + \phi)$ and $\cos(\Theta(t) + \phi)$ terms respectively. In which:

$$\Theta(t) = \int_0^t \left| \frac{d\theta(\tau)}{d\tau} \right| d\tau \quad (7)$$

And $\theta(t)$ is the angular position of the screw shaft at time t . The reason behind this formulation is to enforce $\Theta(t)$ to monotonically increase with time even if the direction of motion changes. This is a necessary condition for the stability of the adaptation scheme. In order to set the phase advance parameter, the phase of the inner closed-loop system was calculated in real-time based on the two-mass model of the ball screw setup, and the angular velocity of the screw shaft calculated from the rotary encoder feedback.

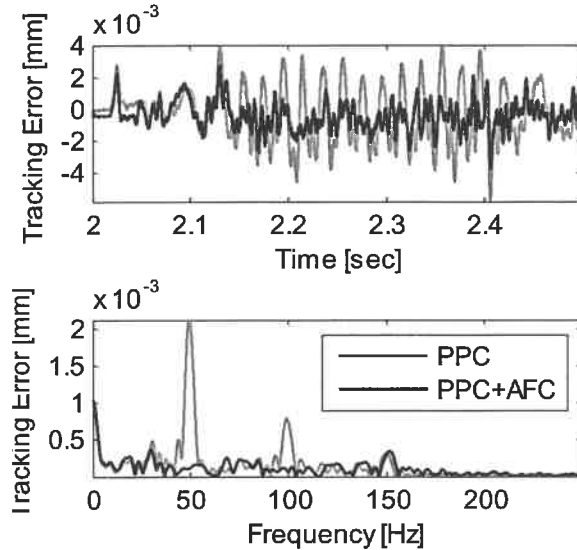


FIGURE 4. Contribution of AFC to the dynamic accuracy of the ball screw drive (a) comparison in time domain, (b) comparison in frequency domain (targeted frequencies: 1/rev & 2/rev).

The results proved that adding AFC to the main controller leads to a significant reduction in the translational tracking errors (Fig. 4).

The same technique was also used to develop a controller to compensate for the effect of periodic cutting forces during machining. Here, it was observed that the frequency content of the

disturbance force was concentrated at the spindle and tooth-passing frequencies (50 and 200 Hz in Fig. 5 and 25 Hz and 100 Hz in Fig. 6) Therefore, these frequencies were targeted for compensation. Again, the experimental results showed the effectiveness of the designed AFC in reducing the translational tracking errors.

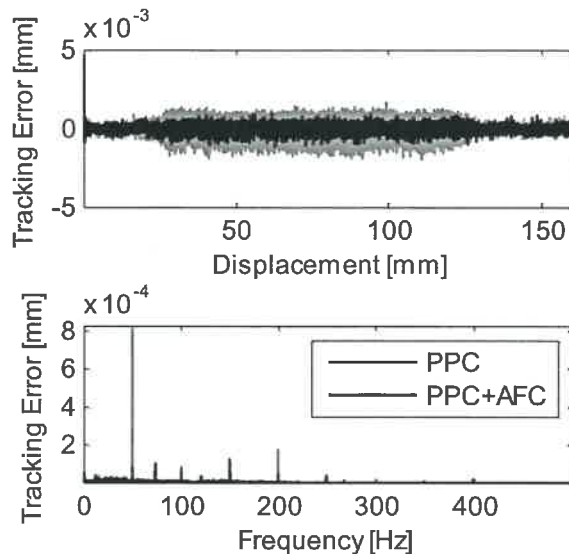


FIGURE 5. Contribution of AFC when machining aluminum 6061 with a 4-flute cutter at 3000 RPM spindle speed and 10 mm/s feed.

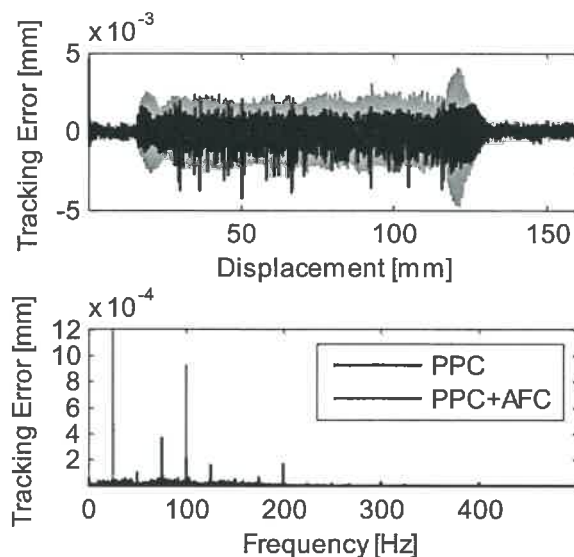


FIGURE 6. Contribution of AFC when machining aluminum 6061 with a 4-flute cutter at 1500 RPM spindle speed and 10 mm/s feed.

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We describe our investigations into the potential use of arrays of electroquasistatic (EQS) and magnetoquasistatic (MQS) impedance sensors as high-resolution inspection and imaging tools. In these designs, arrays of individually addressable EQS sensors can provide a new class of fast-scanning imaging tools to aid in the quality control of for example integrated circuits and photolithographic reticles. We have conducted a number of case studies that evaluate the sensitivity of EQS sensors in detecting both surface and subsurface features and contaminant objects that are typical in an integrated circuit or photomask. There also exists an entire dual class of magnetoquasistatic (MQS) sensors. MQS sensors are particularly advantaged over EQS sensors when detecting features in highly conducting substrates due to the capabilities of magnetic fields to see beneath the surface of conductors at very low frequencies. We have also conducted experiments with centimeter-scale electrodes which demonstrate the effectiveness of such EQS probes.

As an example, we envision using an array of electrodes with dimensions on the order of 100 nm, which are driven with GHz excitation in order to obtain video bandwidth imaging of a substrate of interest. A possible configuration is shown below in Figure 1, along with a variable geometry trench such as might be fabricated on an integrated circuit die. In this configuration, the goal is to determine the dimensions (depth and width) of a sub-micrometer scale trench in doped silicon based upon a measured transimpedance between two sensors in a coplanar array that is scanned laterally over the surface of the silicon. Here the electrodes are 100 nm wide and 50 nm deep. Figure 2 shows the resulting electric field pattern. We envision an array of hundreds or thousands of independent electrodes extending out of the plane of the figure. This will allow simultaneous imaging on hundreds or thousands of channels simultaneously. Given that the electrodes can be excited with frequencies on the order of GHz, we expect to be able to achieve 10 MHz imaging bandwidth on each electrode set, allowing video rate imaging of thousands of pixels simultaneously.

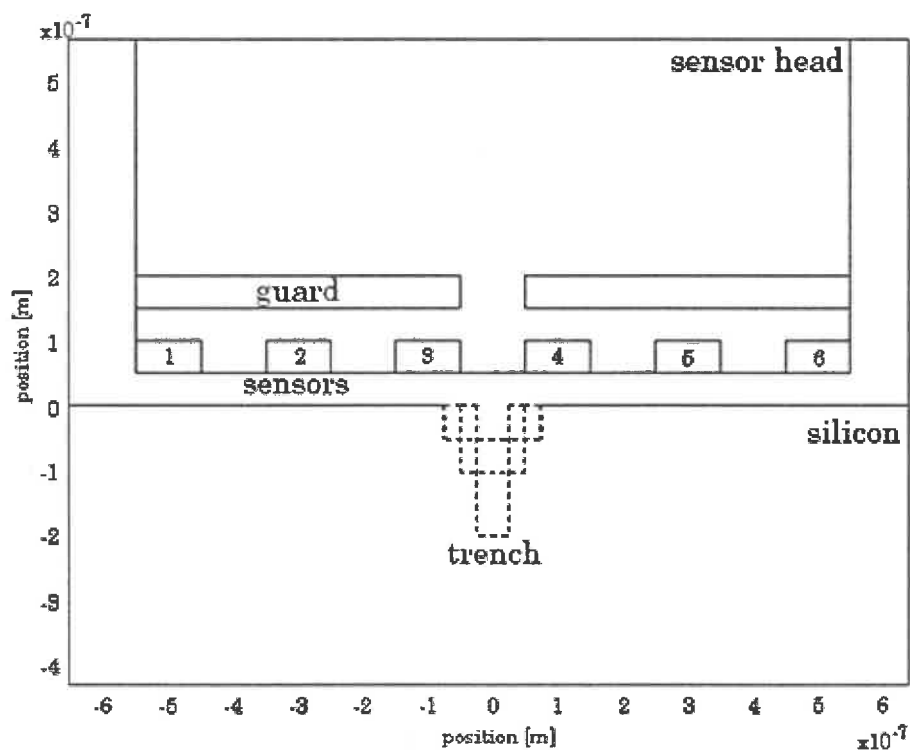


Figure 1: Geometry of electrode and substrate for nanoimager. Electrodes and guard on left (1,2,3) are driven at equal potential, which is the negative of the potential drive for electrodes and guard on the right (5,6,7).

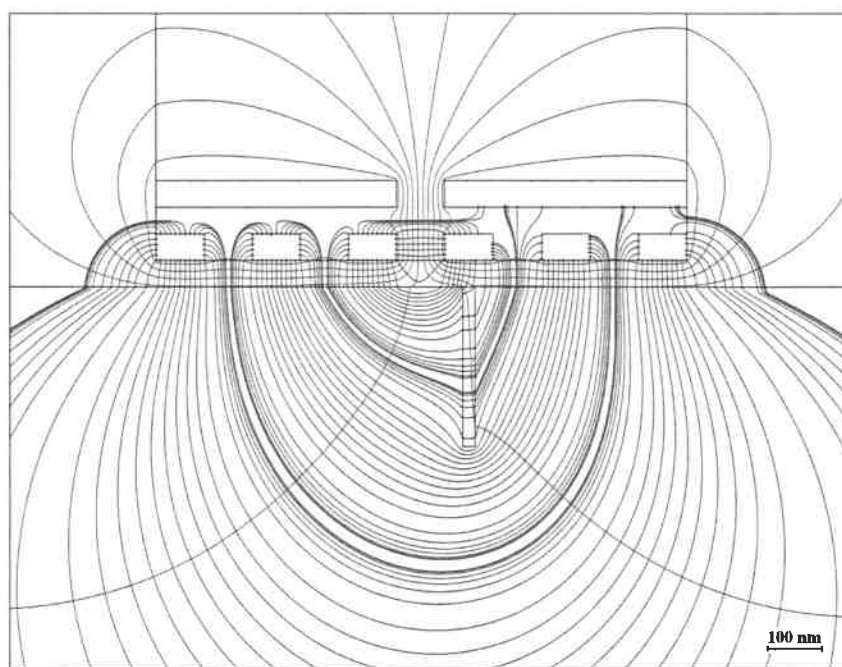


Figure 2: Electric field pattern created in silicon substrate with trench.

A NOVEL LONG-STROKE PLANAR MOTOR

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INTRODUCTION

Planar motion stages (XY tables) are widely used in various manufacturing, inspection and assembling processes. The most common solution to achieve planar (XY) motion is to stack two linear stages together via connecting bearings. The ideal solution is instead to have a single moving element capable of XY motion, eliminating additional bearings; such a stage would also potentially be able to achieve at least some Z motion. There are many efforts towards this ideal motor concept. Sawyer motors are well known for their versatile XY capability, but suffer from inherent cyclic errors as well as constraints imposed by the umbilical cord. Kim and Trumper designed a levitating planar motor with a stroke of $50 \times 50 \times 0.4 \text{ mm}^3$ [1] using Halbach array and phase current commutation to generate both driving and levitating forces [2]. This idea has now been widely used in planar motor designs. In order to extend planar stroke beyond a small fraction of the moving stage size, there have been several long-stroke moving-magnet planar motor concepts in the last 10 years [3][4][5], which are essentially based on a 2-D Halbach array in combination with a 2-D array of coils. These 2-D Halbach magnet motors can potentially achieve very long stroke, but suffer from several issues, such as system complexity, poor coil stacking factor, coil end effects, actuating force shift with stage position, and so on. In this paper, we present a new long-stroke planar motor design with the following advantages: (1) the number of coils linearly increases with the normalized stroke, and thus this design has very good scalability for large stroke applications; (2) the coil terminals exit at the perimeter of the stator, greatly simplifying motor architecture; (3) the coil filling factor can be increased to 70% and thus higher force density is achievable; and (4) it is easy to control/commutate and has superior force linearity due to being free of coil end effects and field edge effects.

LONG-STROKE PLANAR MOTOR PRINCIPLE

Fig. 1 shows the concept of this novel long-stroke planar motor. The motor is composed of a

stationary planar coil array and a moving stage carrying magnet arrays. The active zones of the coil array will track the moving stage. The X1/X2 magnet arrays interacting with the corresponding coils generate forces in Y and Z directions; the Y1/Y2 magnets generate forces in X and Z. These forces enable control of the moving stage motion in 6 degrees-of-freedom.

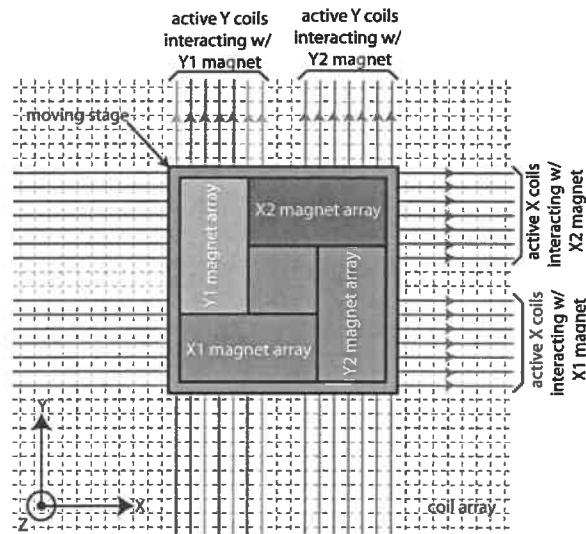


FIGURE 1: Long-Stroke Planar Motor Concept.

Planar Coil Board

The planar coil is made of multiple planar coil boards (PCB). This board can easily be manufactured using standard printed circuit board technology. As shown in Fig. 2, the X and Y coils carry currents in X and Y directions, respectively. Insulator layers (FR4 core or prepreg) are inserted between coils. Up to 16 or more layers of coils can be easily stacked together in a single board using existing technologies. The coils in the same direction at different layers can be connected in parallel or serially, depending on the via design and end termination. There are several advantages in such a coil design:

- **Current Capacity**

Each coil layer thickness can be made from 6 oz copper (200 μm thick) or more. Essentially, each

coil is in the shape of a flat strip, which provides very good thermal conductivity due to the high ratio of surface area to volume. Our test confirmed that laminated copper can carry a sustained current density of 10 A/mm^2 with a 50°C temperature rise above ambient without using an active heat sink. Another advantage of the planar coil board is that the naturally stratified conductor makes it ideally suitable for carrying AC current, because the self-generated alternating magnetic field can easily penetrate the conductor through top and bottom surfaces but produces only low eddy current generation.

- **Lateral Scaling**

Multiple PCBs are aligned side by side in both X and Y directions (similar to floor tiles) to cover the whole moving area. The board-to-board lateral connections (in X and Y) are made at the edges only by connecting pads or through-holes of adjacent boards. As a result, all the end terminals of the entire planar coil array are at the outermost perimeter for ease of wiring to the drive electronics. This allows the motor to be easily extended in both X and Y directions for various applications. The total number of coils increases linearly with the motor stroke instead of quadratically (as in racetrack coil designs).

- **Vertical Scaling**

A single planar coil board can easily be made up to 5 mm thick using available printed circuit board technology. When thicker coils are required for heavy-duty applications, multiple PCBs can be stacked vertically in the Z direction.

- **Distributed Sensor Integration**

Another benefit of using printed circuit boards is the possibility of deploying large numbers of low-profile sensors (such as Hall-effect or capacitance sensors) directly on the board using daisy chain connections.

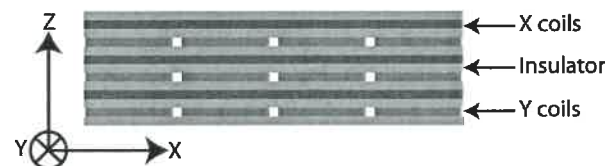


FIGURE 2: Planar Coil Board made from printed circuit board manufacturing technology.

Symmetric Halbach Array

Generally, advantages of a Halbach array over an array of only vertically magnetized elements include: 40% higher magnetic field strength and free of higher order harmonic field (3^{rd} , 7^{th} , and

all even harmonics) as shown in [2]. In this paper, we discuss several additional key features required in Halbach array design for improving force decoupling and linearity in planar motor applications:

(a) the magnet array width should be in integer multiples of its spatial period λ ;

(b) the array magnetization $\vec{M}$ should be mirror-symmetric about its vertical middle axis;

(c) the spacing between two magnet arrays in the same direction should be set at $k\lambda/6 + \lambda/12$, (where k is an integer).

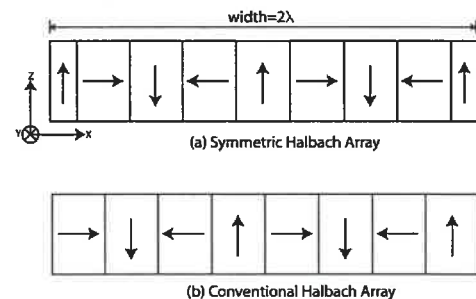


FIGURE 3: 1-D Halbach Magnet Array for Y1.

Fig. 3a shows a cross-sectional view of an example Y1 magnetic array. This is a 1-D Halbach array with uniform magnetization $\vec{M}$ in the Y dimension, i.e. $\partial\vec{M}/\partial y = 0$. Arrays Y2, X1, X2 in Fig. 1 are similarly designed. Physically decoupled actuating forces are always preferred in actuator design. In this planar motor, the Y1 magnet array should produce zero force from its interaction with active coils for magnet arrays X1, X2, and Y2. The decoupling between Y1 and Y2 can be easily achieved by spacing Y1 and Y2 such that the Y1 magnet array is at least $\lambda/2$ away from the active Y coils for the Y2 magnet so that the fringing field seen by Y1 is less than 1% of the magnet remanence. However, some active X coils for X1 and X2 magnet arrays are always right below the Y1 magnet, as shown in Fig. 1. When the magnet array width is an integer multiple of its spatial period, the net force between the Y1 magnets and any X coil is zero. When the array magnetization in Z is mirror-symmetric as demonstrated in Fig. 3a, the net interaction torque is zero as well. Therefore, a non-symmetric design (as shown in Fig. 3b) should be avoided, as its coupling with X coils produces disturbance torques. The 5^{th} order field harmonics produced by the Halbach array will

produce 6th order disturbance forces when using a sinusoidal commutation law. To further reduce the disturbance forces/torques, the spacing between two magnet arrays in the same direction should be set at $k\lambda/6 + \lambda/12$, where k is an integer number.

Field Folding

Usually the magnetic field of a Halbach array is assumed to be primarily sinusoidal with a small amount of 5th order harmonic distortion. This is true at locations inside a long multi-period array but, as the edge of the magnetic field is approached, the field is far from sinusoidal, particularly when the magnet array is only 1- 2 periods wide as Fig. 4a. Designing a commutation law to achieve linear force characteristics for such a non-sinusoidal field is not straight forward. One way to minimize the fringing field effects is to increase the number of periods in the array [1]. Although the disturbance force resulting from the fringing field is unchanged with the increased array width, its effect relative to the total force magnitude is reduced. Another solution is to excite coils only far away from the magnet array edges as in most moving coil linear motors [4], but this will sacrifice force generation capacity. This paper presents a new method to perfectly eliminate the fringing field effect on force generation, which we term field folding.

Using spatial convolution theory, it can be proven that the forces (both lateral and vertical) between a magnet array of width W_m and a current array of period W_m are of identical magnitudes to those between an infinitely wide magnet array with period W_m and a single coil. This principle is illustrated in Fig. 4. A single coil is below a short magnet array in Fig. 4a, producing non-sinusoidal forces due to fringe fields. In Fig 4b, by adding extra coil current (in green) to form a current array of period W_m , the resulting force becomes sinusoidal, the same as that between the single coil current and an infinite-wide periodic array as shown in Fig. 4c. Practically, we only need to excite extra coils up to $\lambda/2$ beyond both sides of the magnet array. This equivalence significantly simplifies the force analysis. Using field folding and standard three phase sinusoidal commutation, the actuating force has excellent linear characteristics, which is highly desirable for high speed precision applications.

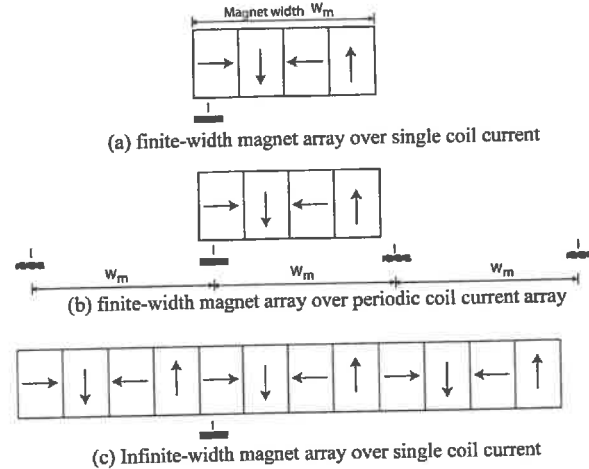


FIGURE 4: Field folding principle.

LONG-STROKE PLANAR MOTOR FORCE ANALYSIS FOR INTUITION-BASED DESIGN

In this section, we present a filter-based method for calculating interacting forces between magnet arrays and the coils. The objective of this analysis is to provide intuitive understanding of the role that each design parameter plays in the overall performance of the motor. The field folding method allows the magnet array width to be considered infinite. The starting point of the analysis is a singular case as shown in Fig. 5a: a Halbach array of infinitesimal height ($\delta \rightarrow 0$) is located at the XOY plane and the magnet remanence is $1/\delta$ so that the overall spatial excitation is unity. As the magnet charges are distributed in the XOY plane only, the magnetic flux below the magnet can be conveniently derived:

$$\vec{B}_\delta = \begin{bmatrix} B_{\delta x} \\ B_{\delta y} \end{bmatrix} = f_M \frac{e^{z/\lambda_c}}{\lambda_c} \begin{bmatrix} -\cos \frac{x}{\lambda_c} \\ \sin \frac{x}{\lambda_c} \end{bmatrix} \quad (1)$$

where $\lambda_c = \lambda/2\pi$ is the characteristic wavelength and $f_M = 0.9$ is the magnet array factor indicating how close the analyzed array is to an ideal continuously varying field Halbach array. This result shows that a 4-segment Halbach array is good enough for motor applications and increasing the number of magnet segments per period has little marginal benefit. Eq. (1) is the impulse response in the spatial domain under 1-D Halbach magnet array excitation. Accordingly, the average flux density over the coil cross-section in Fig. 5(b) can be derived as the convolution of Eq. (1) with several spatial filters resulting from magnet height H , coil

width W , and coil thickness T as illustrated in Fig. 5c. As a result, the motor achievable acceleration can be represented as:

$$a = \frac{JB_r}{\rho_m} f_M f_H f_C f_G f_W f_T f_S$$

where ρ_m is the magnet mass density, JB_r/ρ_m represents the acceleration limit of moving-magnet motors (note that that the acceleration limit for moving-coil motors is JB_r/ρ_c , where ρ_c is coil mass density), and all f gains are decoupled dimensionless factors representing effects of various design parameters on overall performance. $f_H = \lambda_c(1 - e^{-H/\lambda_c})/H$ is the magnet height factor, indicating H should be designed at near λ_c for maximum load capacity without sacrificing acceleration; $f_C = 0.5$ is the factor of sinusoidal commutation; $f_G = e^{-G/\lambda_c}$ is the air gap factor; $f_W = \text{sinc}(W/2\lambda_c) = 3/\pi$ is the coil width factor; $f_T = 1 - e^{-T/\lambda_c}$ is the coil thickness factor, indicating the PCB thickness should be set around λ_c for energy efficiency; and f_S is the stacking factor (the ratio of the volume of force generating coils to the volume of the entire PCB). These scaling factors provide intuition on how to set the design parameters of the presented motor. For other types of motors, similar results can be derived in this way.

STATUS AND OUTLOOK

A planar motor prototype is being constructed, as shown in Fig. 6. The key design parameters and predicted performance are listed in Table 1.

Table 1. Prototype Planar Motor Parameters

Parameter	Value
Stroke X/Y/Z, mm	700/250/4
Armature L/W/H, mm	185/185/62
Total moving mass, kg	2.2
PCB L/W/T, mm	559/320/5.7
PCB layer number	16
Current operation density, A/mm ²	10(cont.), 15(peak)
Coil thickness/layer, mm	0.213
Prepreg thickness/layer, mm	0.127
Stacking factor	30%
Magnet remanence flux, T	1.325
Magnet array period λ , mm	30
Magnet height H , mm	7.5
Magnet length, mm	120
Peak acceleration a , g	5(cont.), 7.5(peak.)

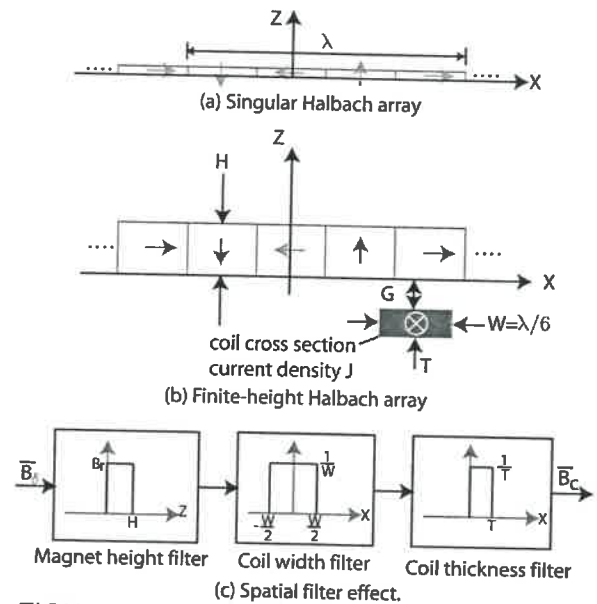


FIGURE 5: Geometric parameters of the coil and the magnet array.

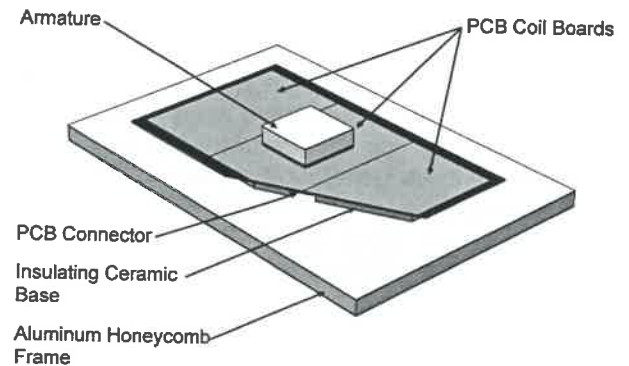


FIGURE 6: Solid model of the long-stroke planar motor prototype.

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INTRINSIC DAMPING OF LINEAR HALBACH BRUSHLESS MOTORS

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INTRODUCTION

Permanent magnet synchronous linear motors (PMSLM) are among the most prevalent in precision machines due to their long-range travel and high precision and accuracy [1]. Some examples of multiple PMSLM driven 6 degree-of-freedom (DoF) stages include the Multi-Scale Alignment and Positioning System (MAPS) [2, 3, 4] of which this paper deals with and the Subatomic Measuring Maching (SAMM) [5]. In this paper we describe a phenomenon, found during system identification of the MAPS system, showing that the linear motors themselves have natural damping due to the electromagnetic interaction between the magnet array and the aluminum mandrel.

LINEAR HALBACH MOTOR

The linear motor, shown in figure 1, is comprised of the coils or stator and magnet array or mover. The magnets are oriented 90° to each other, ver-

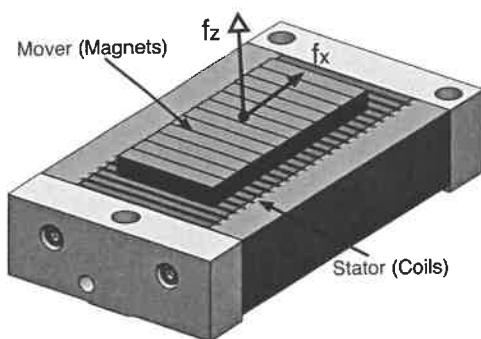


FIGURE 1. Linear-Motor Components

sus the standard 180°, making it a Halbach array. The Halbach array produces a levitation force independent of the thrust force, where standard array produces only thrust. Klaus Halbach invented the Halbach array in 1980 for use in particle acceleration [6]. Figure 2 shows an exploded view

of the linear motor revealing the aluminum mandrel of which the coils are wrapped around.

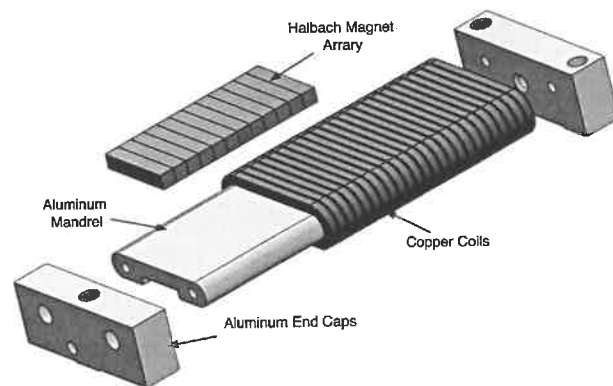


FIGURE 2. Linear-motor exploded view

SINGLE MOTOR EDDY-CURRENT DAMPING

It was observed during system identification of the MAPS system that the linear motors themselves produce eddy-current damping only in the axial direction of the motor. This was assumed to be caused by the interaction between the magnet array and the aluminum mandrel, of which the coils are wrapped around (see figure 2). To differentiate between the motor damping effect and other possible damping sources, such as the air bearing, a spare magnet array and mandrel were studied removed from the MAPS system.

The experiment was conducted on a Monarch VMC45 vertical machining center with the magnet array carried in the machine quill and the mandrel mounted on a Dover linear air bearing which was in turn mount on the Monarch table. The force was measured with a Chatillon DFA-R-ND force gage with a Remote-10 remote load cell (50 N capacity and 0.005 N resolution). The setup is shown in figure 3. The experimental data, and fil-

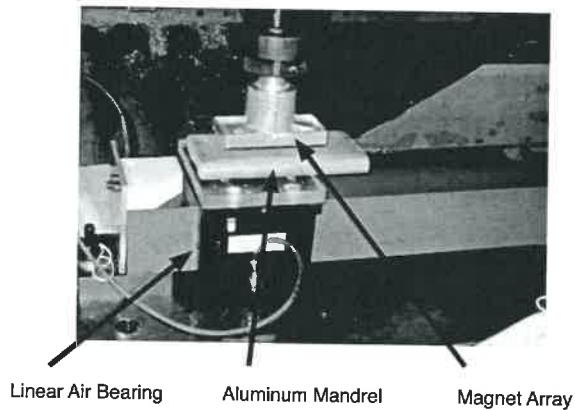


FIGURE 3. Experimental setup to study the interaction between Halbach magnet array and aluminum mandrel

tered data, are shown in figure 4. The results are

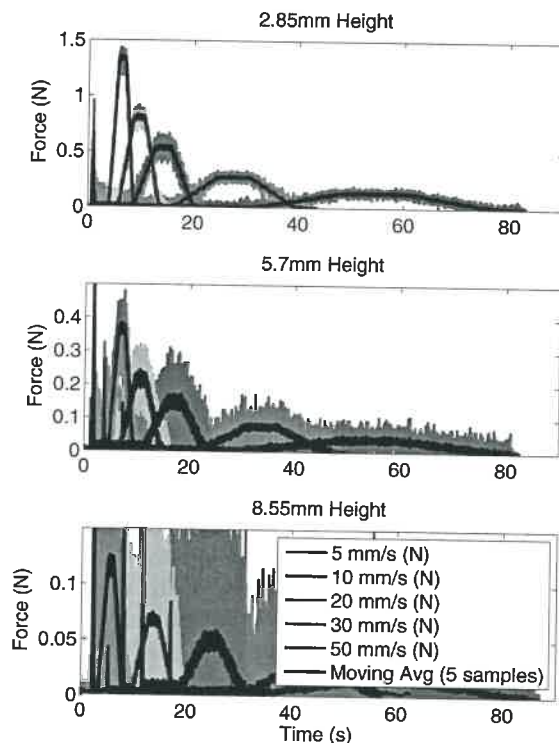


FIGURE 4. Experimental force versus time as the aluminum mandrel passes by the magnet array at various speeds and separation heights

presented in figures 5 and 6 where it is shown that the damping force is linearly dependent on velocity and increases as gap height decreases.

An inverse squares curve fit of damping coefficient

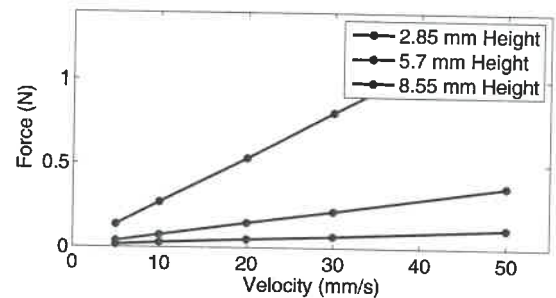


FIGURE 5. Eddy-current force versus mandrel velocity

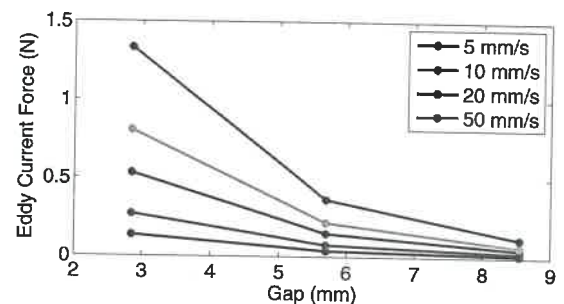


FIGURE 6. Eddy-current force versus gap separation h

cient versus gap height is presented in figure 7. Since the platen only has a range of micrometers in z , the gap between magnet array and mandrel stays around the nominal distance of 4.83 mm (the thickness of the coils), we estimate the damping coefficient for MAPS' motors to be (see figure 7):

$$B_m = 9.4 \text{ N/(m/s)} \quad (1)$$

Interestingly, it was also noticed during identi-

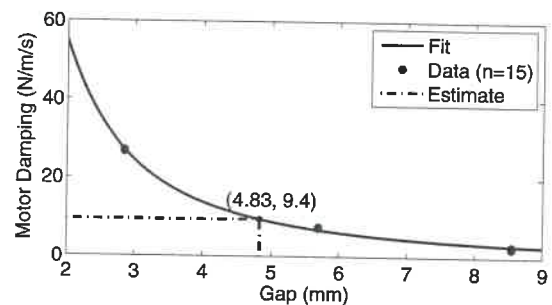


FIGURE 7. Motor damping data, measured removed from MAPS, and an estimate for a gap of 4.83 mm between magnet array and mandrel

fication that the motor damping only acts if the

platen is driven along the motor's axis. This was verified by conducting the same experiment with the magnet array rotated 90 degrees as the mandrel passed under. A comparison of the rotated and not rotated data, for the smallest separation gap and thus the largest force, is shown in figure 8. These results verify the observation that

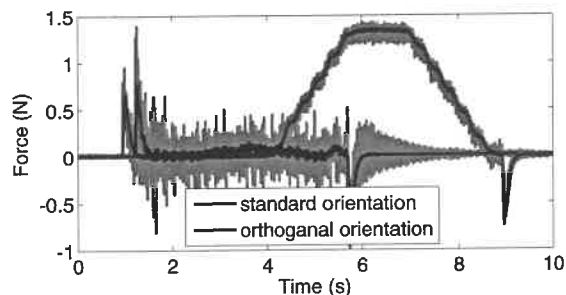


FIGURE 8. Comparing experimental damping results, both at a mandrel-magnet array separation gap of 2.85 mm, of the magnet array aligned and rotated by 90 degrees relative to the mandrel motion

the eddy-current damping produced by the interaction between the Halbach magnet array and the aluminum mandrel only acts along the axis of the motor.

SYSTEM DAMPING

In this section we compare the estimated motor damping coefficient found in the previous section to coefficient found through system identification of MAPS as a system (see [4] for more details on the system identification). The linear motors, for MAPS, drive the platen, or MAPS' mover, as shown in figure 9. Notice that motors #1 and #3 drive the platen primarily in the x-direction and #2 and #4 drive the platen in the y-direction. The platen is levitated on a novel porous air bearing (see figure 10). The air bearing allows non-contact motion of the platen and consists of two lands, for levitation pressure, and two vacuum grooves. Vacuum is for the purposes of preloading the bearing and other process related tasks. The pressure supplied to the air bearing for levitation was 60 psi.

By the results of the last section, namely that damping occurs only in the axial direction of the motors, travel in the x direction incurs damping only from motors #1 and #3 while damping from all four motors will act on the platen rotating about

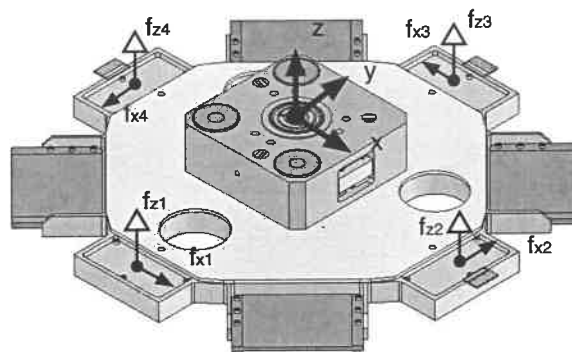


FIGURE 9. The 8 local motor forces $[f_{x1} f_{z1} \dots f_{x4} f_{z4}]$ drive the platen in all six degrees of freedom

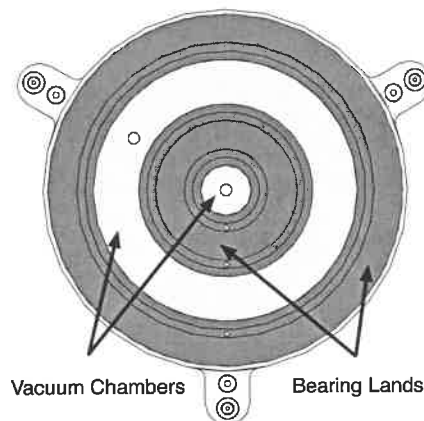


FIGURE 10. The air bearing, consisting of two lands, for levitation pressure, and two vacuum grooves for process related tasks, allows non-contact motion of the platen

the z -axis giving the following linearized equations of motion:

$$M\ddot{x} + 2B_m\dot{x} = F_x \quad (2)$$

$$I_z\ddot{\theta}_z + 4R^2B_m\dot{\theta}_z = T_z \quad (3)$$

Damping coefficients were calculated for the x , y , and θ_z directions, over the operating range of the air bearing vacuum of 0-450 mmHg. Assuming that the single motor damping coefficient B_m obeys the equations of motion given above, the damping was estimated and is shown in figure 11. It should be noted that the motor damping

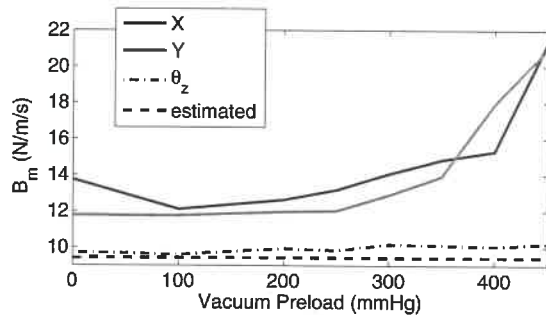


FIGURE 11. Motor damping B_m versus the vacuum preload of the air bearing showing that the air bearing affects the damping in x and y but not in θ_z

coefficient is very close to the estimate of 9.4 N/(m/s) when found by the θ_z dynamics. On the other hand when calculated by the x or y dynamics, damping increases as the vacuum preload increases. It is obvious by figure 12, showing system response to non-zero initial conditions in x , y , and θ_z , that the air bearing is adding system damping to the already present eddy-current damping of the motors as vacuum preload increases. It is curious, and calls for more analysis,

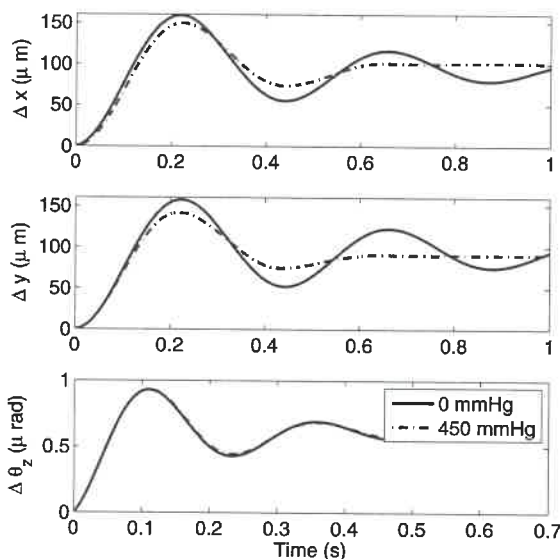


FIGURE 12. System response to non-zero initial condition in x , y , and θ_z , where vacuum preload influences the system damping only in x and y that the air bearing adds no system damping in the θ_z direction.

CONCLUSION

This paper presented experimental results verifying that the Halbach linear motors, with an aluminum mandrel, produce eddy-current damping in the axial direction only. System damping is increased only in the x and y directions due to the vacuum preloaded air bearing. When damping is not affected by the preloaded air bearing, as in the θ_z direction, the motor damping calculated matches the damping estimated by the data collected from a single magnet array and mandrel removed from MAPS.

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HYDROSTATICS ON A HALF SHELL

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INTRODUCTION

Hydrostatic bearings are traditionally associated with high precision applications, and they are comprised of precision finished elements. This paper presents the design of a self-compensating hydrostatic bearing made from two halves so it can be assembled about a shaft, and the bearing surfaces are made from plastic or rubber bonded to half shell structures. The bearing bore structure can have a precision shape but rough surface finish, or the bearing bore can be as-cast and then the bearing surface vacuum-held by a master cylinder can be bonded in place. The bearing geometry can be molded into the bearing surface, so the entire system can be made for low cost.

The self compensating features are unique in that they are not located on opposite sides of the shaft, yet the initial design has an efficiency of 23%; hence the load capacity can be calculated by multiplying the efficiency by the projected area and the supply pressure, or $0.23 \times \text{Length} \times \text{Diameter} \times \text{Supply Pressure}$. The bearing can be run with any type of fluid including water. In addition, the new self compensating features are conducive to hydrodynamic operation when the shaft speed is sufficient; thus pump power can be greatly reduced once a minimum shaft speed is attained. Tests on a 100 mm diameter shaft confirm the design theory.

DESIGN

Initial designs were laid out in CAD based on previous work [1]. The designs were analyzed as a fluid resistance network as outlined in [2]. The fluid resistance network analysis was used to complete a first order design optimization. Figure 1 shows the design of the bearing. Figure 2 shows how the bearing would look when rolled into its final state. This flat pattern is designed to cover a 165° arc as opposed to a full 360° journal bearing. The bearing is designed for a shaft that is nominally 100 mm in diameter and has three pocket regions to give the system some tilt stiffness. The dimensions were sized so that the bearing length is roughly twice the diameter.

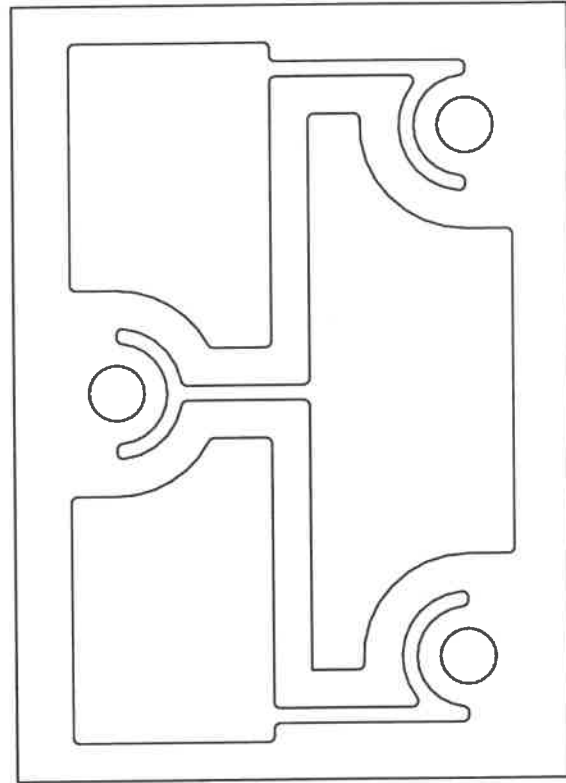


FIGURE 1. Hydrostatic bearing flat pattern

All the fluid routing aside from the fluid inlets is accomplished by the surface features. The lack of additional hoses and orifices increases robustness against plugging and biofouling and allows water to be used as a working fluid.

BUILD AND TEST

The hydrostatic bearing features were machined into a nominally 2.54 mm thick sheet of adhesive-backed ultra high molecular weight high density polyethylene. An aluminum housing was machined out of a solid block of aluminum to prevent any warping due to residual stresses as occurred in initial efforts to use a tube cut in half axially. A shaft was machined to provide 0.13 mm nominal gap. The aluminum housing, plastic and shaft were heated to 130° C in a furnace with a 127 µm thick sheet of shim stock between the shaft

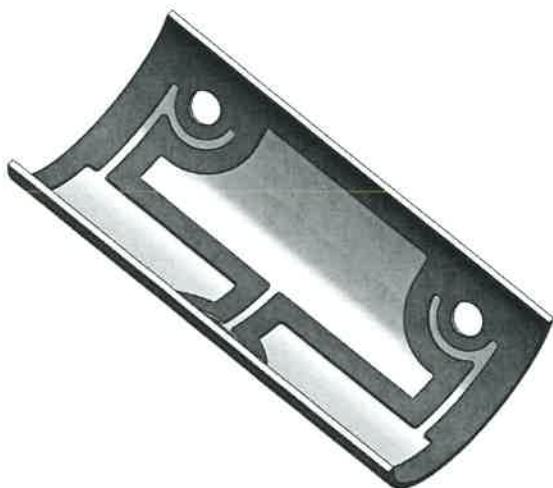


FIGURE 2. CAD model of the hydrostatic bearing in its formed state

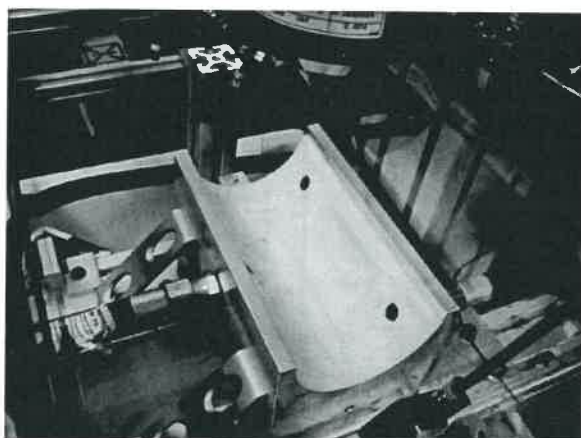


FIGURE 3. Hydrostatic bearing in test setup

and the plastic to thermally form the plastic to the proper gap, where the bearing gap would ideally be the thickness of the shim stock. This process reduced the stresses in the plastic and reduces issues with delamination of the plastic. A test setup was built for the bearing which includes a dedicated impeller pump, filter, flow meter, digital pressure readout, and ball valves for flow control in the fluid circuit. Air bearings are used to constrain the shaft axially. An Admet¹ 5604 single column universal testing machine and 300 lb load cell are used for applying load to the shaft. The 5604 is driven by Admet's MTestQuattro software. Lion Precision² U3B eddy current probes driven

¹www.admet.com

²www.lionprecision.com

by an ECL202 driver measure the shaft location. The MTestQuattro system records readings from the load cell, eddy current probes, and pressure gauge. Figure 3 shows the plastic glued into the aluminum housing sitting in the test rig.

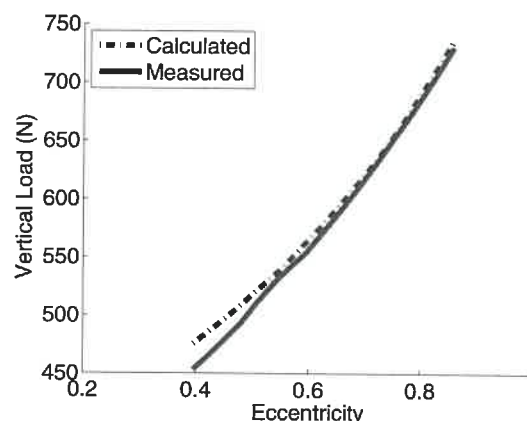


FIGURE 4. Comparison of calculated and measured load

The results from the bearing design are promising. Figure 4 shows the calculated and measured vertical load that the bearing supports. The calculated values come from a Matlab script solving the hydraulic resistance network model for the bearing. Figure 5 shows the stiffness of the bearing, where the stiffness is defined as the change in force divided by the change in bearing gap, $k = \Delta F / \Delta h$. These two data sets demonstrate that the design is deterministic and that the hydraulic resistance model reliably predicts the performance of the bearing. Figure 6 shows the measured efficiency of the model, where the efficiency is the load divided by the supply pressure multiplied by the projected area of the bearing, $\bar{F} = F / PDL$, where D and L are the bearing diameter and length, respectively. At 75% gap closure, the bearing has a 23% efficiency.

Analysis was conducted to determine the effect of wrapping the groove further around the inlet holes. Figure 7 shows an example of increasing the amount the collector wraps around the inlet as well as the angle used to measure the additional length. The grooves were changed symmetrically, and the angle, θ , is half the total amount of additional arc. Figure 8 shows the results of an analysis done for the same bearing with different additional angles of additional groove wrapping around the inlet. The curves shown are for the bearing operating at 15 psi inlet pressure. As

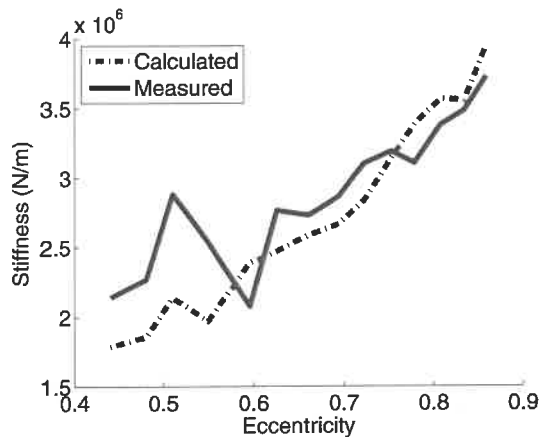


FIGURE 5. Comparison of calculated and measured stiffness

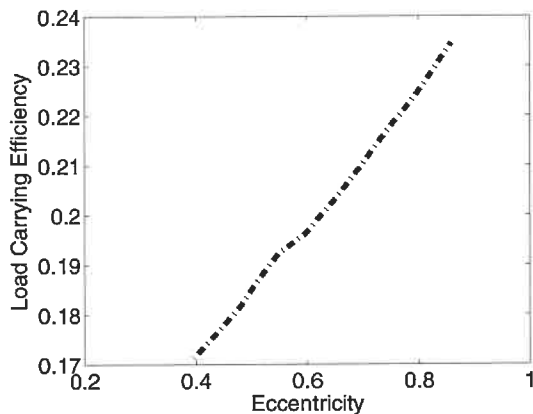


FIGURE 6. Load carrying efficiency

the collector wraps further around the inlet, the amount of vertical load increases. However, the additional load capacity comes at the cost of additional flow rate. Figure 9 shows how the specific flow rate increases with increasing angle of additional wrap. Specific flow rate is defined as $\bar{Q} = Q / (P\pi D h_o^3 / 12\mu L)$, where Q is the total flow rate and h_o is the bearing gap when the axes of the bearing and shaft are aligned.

Another analysis was done to examine the effect of the collector land thickness, T , shown in Figure 7. This land can be adjusted to change the fluid resistance of the collector and thus the resistance ratio, which is defined as the ratio of the resistance of the inlet to the resistance of the outlet. Figure 10 shows the results. The abscissa is the resistance ratio at an eccentricity of 0.01 averaged over the three pockets. The initial stiff-

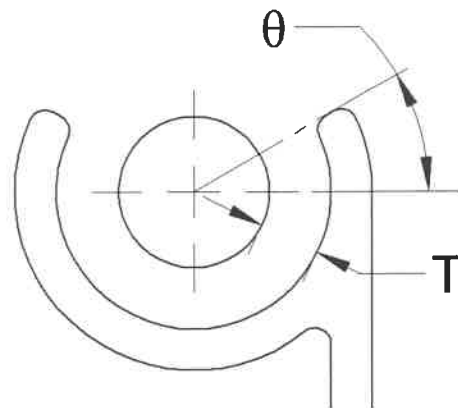


FIGURE 7. Diagram of collector

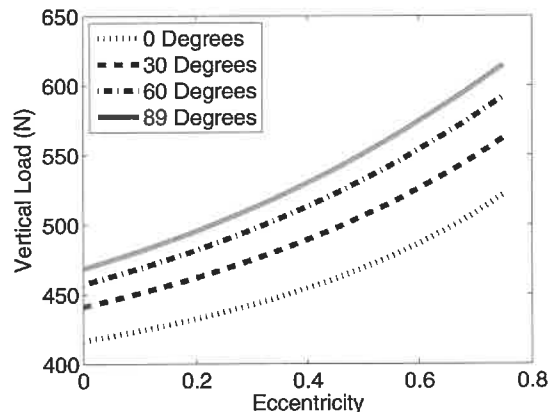


FIGURE 8. Load sensitivity to additional collector land

ness is calculated as a change in vertical force for a change in eccentricity of 0.01 to 0.02. The reported load carrying efficiency is calculated at 75% gap closure. As can be seen, varying the resistance ratio trades load carrying efficiency for stiffness, and both cannot be optimized simultaneously by means of the resistance ratio alone.

CONCLUSION

This work demonstrates the deterministic design theory behind a hydrostatic bearing covering less than 180° of the shaft surface. This paper evaluates just one half of a bearing to support a horizontal heavy shaft. This bearing design facilitates installation and repair of hydrostatic bearings in support of large shafts. Future work will investigate the hydrodynamic performance of this type of bearing. The ultimate goal is to be able to produce a low cost bearing which runs hydrostatically

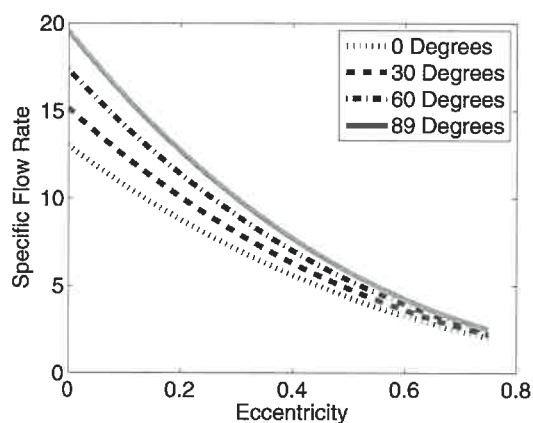


FIGURE 9. Flow sensitivity to additional collector land

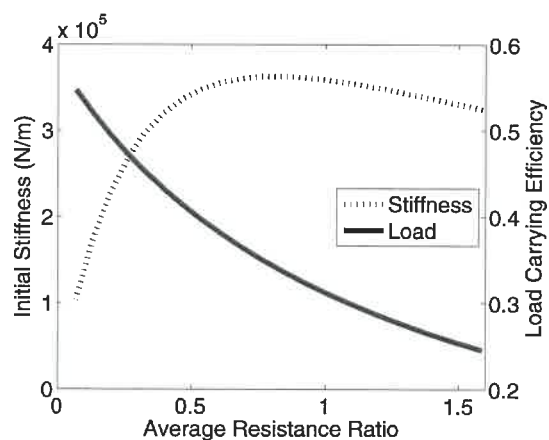


FIGURE 10. Bearing performance sensitivity to resistance ratio

at low shaft speeds and hydrodynamically at high shaft speeds to reduce operating cost by allowing pumping power to be reduced.

ACKNOWLEDGEMENTS

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SPATIAL MICRO MANIPULATION WITH THIN WIRES AND PIEZO ELEMENTS

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INTRODUCTION

For these year, micro-nano manipulation is one of well-known techniques to operate such bio-cell and micro assembly. So several fine mechanisms with precise guides and motor have been developed. The piezo driven stages also have been proposed and nanometer positioning resolution has been achieved. When these positioning devices are implemented into the equipments, then the size of these mechanical structures should be much smaller as the space is so limited to be installed.

In this report, the precise manipulation that the target is connected with three thin wires and they are driven by the piezo elements with the cantilevers is discussed. At first, the spatial trajectory of the target is given by the simulation based on the simple geometrical model. When the displacements of three piezo elements are given to the equations, then the location of the target can be decided since these forces should be balanced. In the primary experiments, the typical positioning performances such as circle and square are checked by the microscopic coordination instrument. The linearity of 0.4% was achieved with the open loop control. The dynamic response has also discussed and the resonance frequency around 80Hz has been measured.

In the conclusions, several results are summarized and the future works including the practical applications as well as the implementation of feedback closed loop are also proposed.

PRINCIPLE OF PRECISE POSITIONING BY THREE THIN WIRES CONNECTED WITH PIEZO ELEMENTS

Three thin wires and piezo elements

In Fig.1, the isometric view of the needle manipulator that is connected with three thin wires and the piezo elements with the cantilevers. The needle position in X-Y plan can be controlled by three piezo displacements as well as it also can be pushed by another piezo elements that is set at the bottom of the needle.

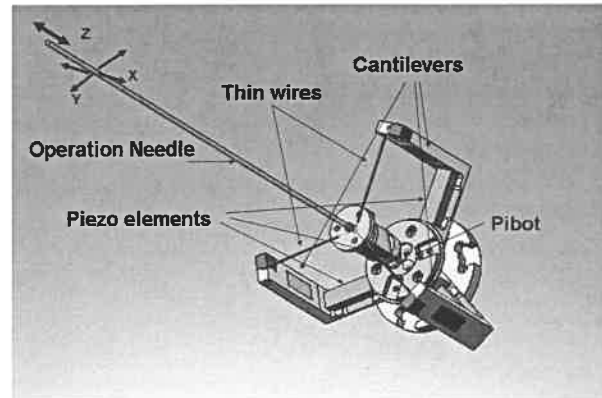


FIGURE.1 Isometric view of needle manipulator connected with three thin wires and piezo elements.

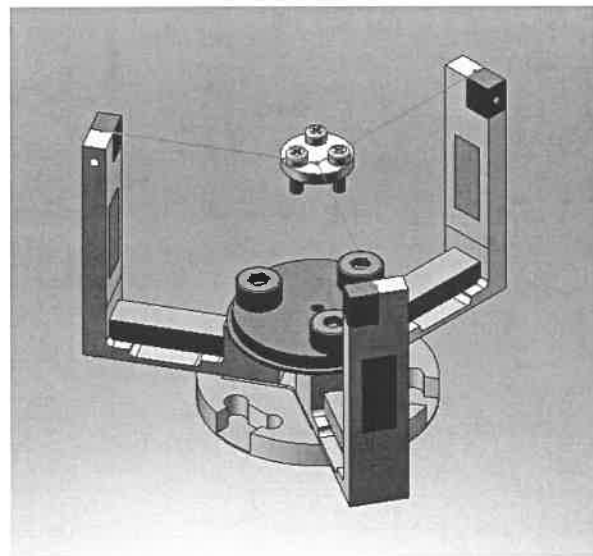


FIGURE.2 .XY position in plane can be controlled by three input displacements of piezo driven cantilevers.

This simple layout can provide the fine positioning within XYZ spatial range to the needle end so that it can manipulate the target precisely. In order to discuss the relation between the XY coordinate position of the central small target and three piezo displacement at first, the simple layout is discussed as shown in Fig.2.

Here the target is connected to the end of the cantilevers driven by the piezo elements with the thin wires. These wires are connected with 120 degree of angle separation. As shown in Fig.3, d_1, d_2 and d_3 are the displacements of these actuators, k_1, k_2 and k_3 are the equivalent spring constant of the cantilevers and the wires, the length of the wire is L . Then XY coordinate position of the small target is given by the static balance of the forces.

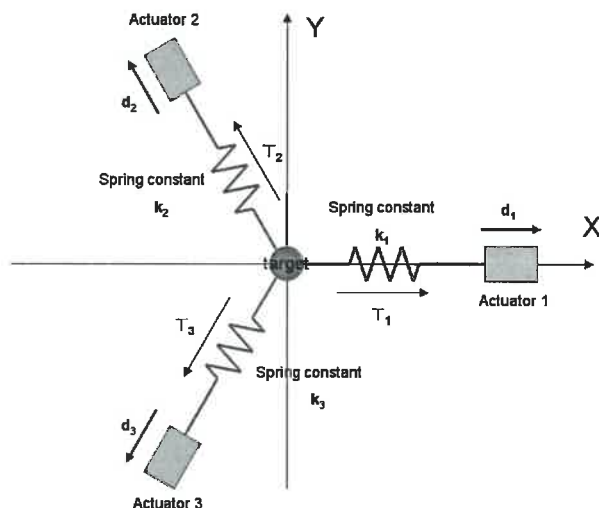


FIGURE.3 Diagram of simple mechanical model that the central small target can be positioned by three thin wires driven by actuators.

$$k_1 \{d_1 - X\} - k_2 \left\{ X + d_2 \cos\left(\frac{\pi}{3}\right) \right\} - k_3 \left\{ X + d_3 \cos\left(\frac{\pi}{3}\right) \right\} = 0$$

$$k_1 \left\{ d_2 \sin\left(\frac{\pi}{6}\right) - Y \right\} - k_3 \left\{ d_3 \sin\left(\frac{\pi}{6}\right) - Y \right\} - k_1 \left\{ d_1 \sin\left(\frac{Y}{L}\right) \right\} = 0$$

Here $L \gg Y$ and $k_1 = k_2 = k_3$, then the target position is given by the following equation.

$$X = \frac{1}{6}(2d_1 - d_2 - d_3), Y = \frac{\sqrt{3}}{4}(d_2 - d_3)$$

Thus the position of the small target that is connected by three thin wires can be decided by the displacements of the actuators.

EXPERIMENTAL SETUP AND RESULTS

Experimental setup

As shown in Fig.4, the primary experimental setup was fabricated and the basic performances were checked when the appropriate input signals are given to each piezo elements. The position of central small target is measured by the microscope and the numerical coordination can be given by the image analyzer.

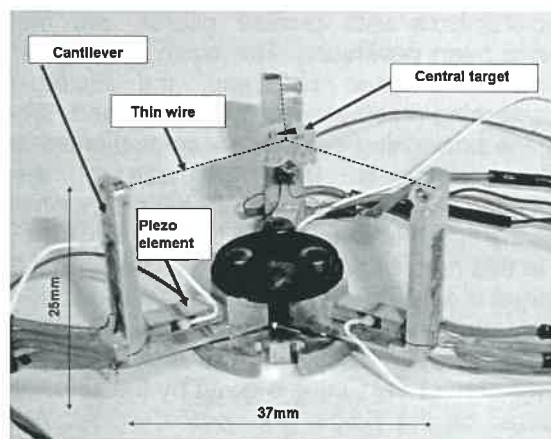


FIGURE.3 Experimental setup that is composed of the central small target and three thin wires connected the cantilevers driven by piezo elements.

Experimental results

In Fig.5, the theoretical and the experimental trajectories are shown when the input signals are given to three piezo elements so that the central target can move along the circle with the diameter of 20μm. In Fig.6, the 5μm square trajectories with the theoretical estimation and the experimental result are shown. It is found that there are the significant errors with the simple open loop control as the piezo elements have non-linear characteristics and the cantilevers are bent definitely. In order to improve such the static performance, the tension force and the bending amount are monitored by the strain gauges that are attached on the both side of the cantilever as shown in Fig.7. These simple sensors can detect the bending amount of the cantilever easily and allow to compensate the errors from such the bending and non-linear performance of the piezo elements. The optical sensor is also available to detect the bending

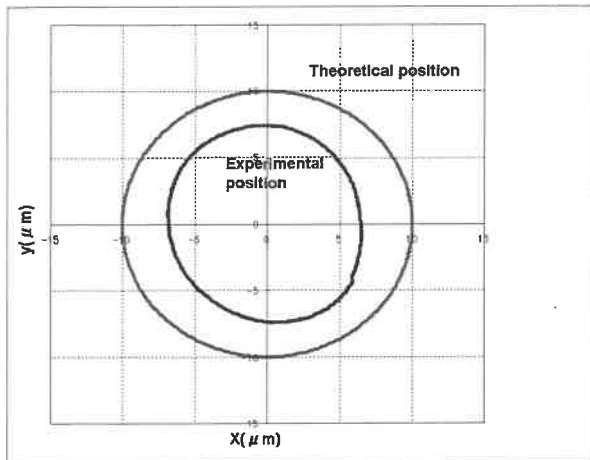


FIGURE.5 Theoretical and experimental trajectories of the small target that is controlled to move along the circle with the diameter of 20μm.

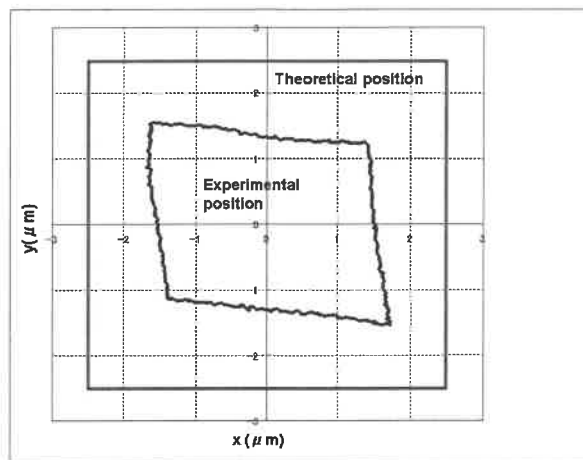


FIGURE.6 Theoretical and experimental trajectories of the small target that is controlled to move along the square of 5μm

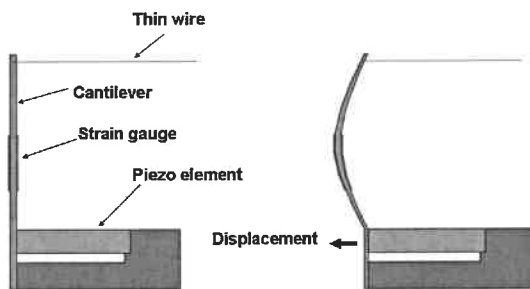


FIGURE.7 Strain gauges are attached on the both side of the cantilever to monitor the bending behavior..

behavior, however, the strain gauge is employed at first to check the possibility of error compensation as well as the closed force feedback property.

In Fig.8, the typical results that can indicate the effect of the simple bending compensation function are shown. When the input signals to the piezo elements are given so that the small wire-connected target can move along the circle trajectory of the diameter of 40μm , then 25% error to the theoretical trajectory appears without compensation. However the error can be decreased to less than 5% with the cantilever bending compensation function. There are still motion error with the trajectory since the linearity of the sensors and the hysteresis of the piezo actuators are still remained.

In Fig.9, the appropriate input signals to three

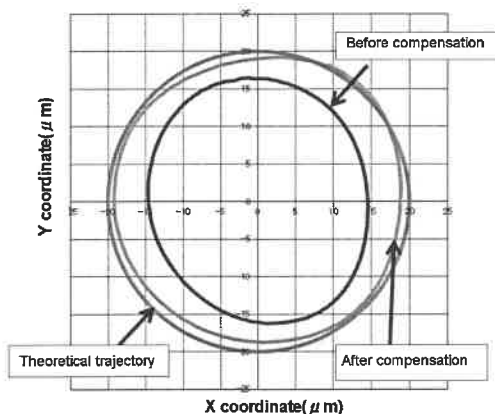


FIGURE.8 Experimental trajectories with and without compensation so that the small target is controlled to move along the diameter of 40μm

piezo actuators are given so that the target can move on Lissajous line. It is found that the target motion could be successfully controlled with the dynamic range of 10Hz and this results can indicate that the small sample such a bio-cell can be easily positioned under the microscope that has a limited space for implementation.

In order to provide such rotational motion with the central sample, another device has been implemented into the central part as shown in Fig.10. Here the central target with the hole can be driven with the frequency of 10Hz to 120Hz. Then the rotary part with the pin shown in Fig.11, that is inserted into the hole of the central target can be rotated differentially.

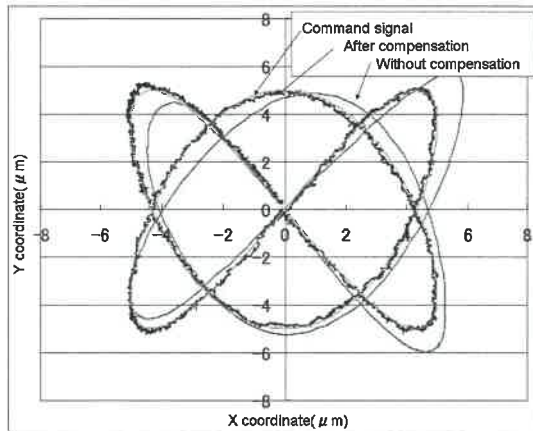


FIGURE.9 Lissajous trajectories can be achieved with the special waveform input so that the target can move on the complicated path precisely

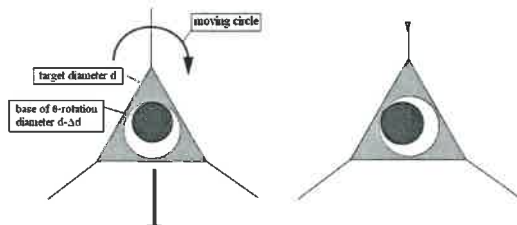


FIGURE.10 Rotary part drive with the differential diameters of the inner pin and the outer hole

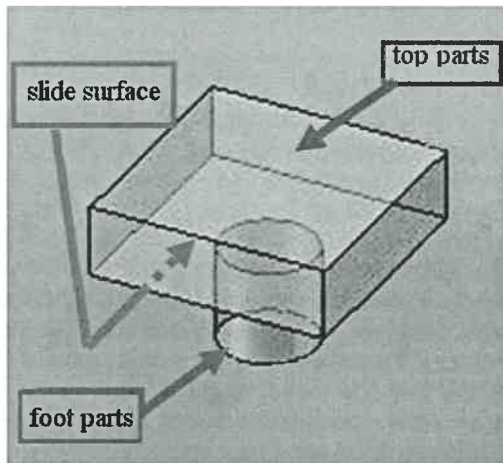


FIGURE.11 Rotary part with the pin that is inserted into the central part with the hole.

This layout can allow the central part to rotate precisely so that such bio sample on it can be easily aligned to any orientation under the microscope.

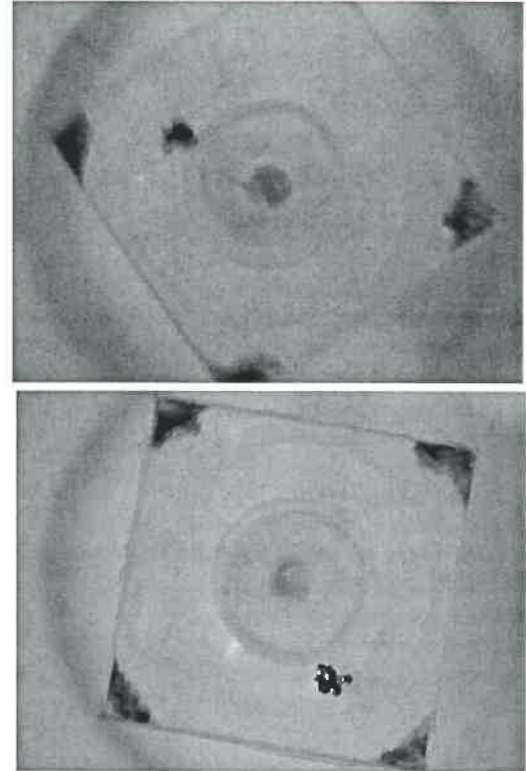


FIGURE.11 Rotary part is rotated with the frequency of 0.25Hz, when the central part is driven by three wired piezo elements with 75Hz.

CONCLUSION & FUTURE WORKS

In this paper, the precise manipulation that the target is connected with three thin wires and driven by the piezo elements connected with the cantilevers. With the open loop positioning control with the cantilever bending compensation function, the motion error less than 5% has been achieved. To improve the accuracy much more and better operability, the closed loop position feedback property and the haptic interface function will be implemented

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Metrology Assisted Manufacturing

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INDUSTRY BACKGROUND

As with most industries, Aerospace is constantly being pushed to drive down production costs, increase quality and increase flexibility. One major way of achieving better quality is through automation, and for a long time Aerospace manufacturers have invested in large, bespoke machinery for the manufacture of parts to a high tolerance, however, these tend to be very expensive and dedicated to a single process (inflexible). With the current trend towards leaner manufacturing, shorter development cycles and more variation in the products on offer, these bespoke machines can be less suitable.

The automotive industry has had to deal with exactly these issues for the past 30 or so years, and their response was to use industrial robots to increase quality, whilst reducing production costs, but still allowing flexibility to produce multiple variants down the same production line. This approach is now being introduced to the aerospace industry, with a rise in robotic applications taking hold across all aerospace sectors.

HIGH ACCURACY ROBOTICS

Industrial robots provide the flexibility and cost effectiveness required, however, they have always been regarded as lacking in the accuracy required to reach typical aerospace tolerances.



Recently, though, advancements have been made in increasing this robotic accuracy both through the robot manufacturers themselves (better design and manufacture) and through the application of external control mechanisms to provide some form of error correction. This

control is generally performed through the use of an integrated metrology system, which could be anything from a 2D vision system, through additional axis encoders, to a complete 6DoF measurement monitoring system.

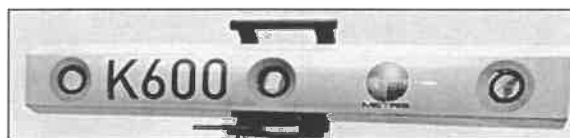


FIGURE 1. Optical Coordinate Measuring Machine

Generally, the difficult part of these applications is to be able to seamlessly integrate the metrology without affecting the benefits of the robot. Nikon Metrology have been working within the robotics industry for over 10 years, and have developed a number of solutions to help increase robot accuracy ranging from simple calibration to fully integrated robotic control systems. The particular solution required can be driven by many factors including tolerances, time, cost, level of automation and the process itself. For the highest tolerance requirements, an integrated solution is preferred and Nikon Metrology's Adaptive Robot Control is such a solution.

UNDERLYING METROLOGY TECHNOLOGY

Adaptive Robot Control (ARC) works by using a photogrammetric camera system, capable of tracking infra red LEDs, to determine the position of a robot tool relative to a fixture or part in 6 degrees of freedom.

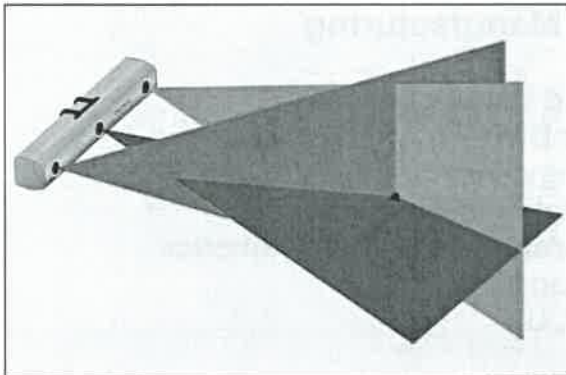


FIGURE 2. LED position triangulated in 3D space

This allows an error from a nominal position to be calculated and sent back to the robot for correction in a simple control loop.

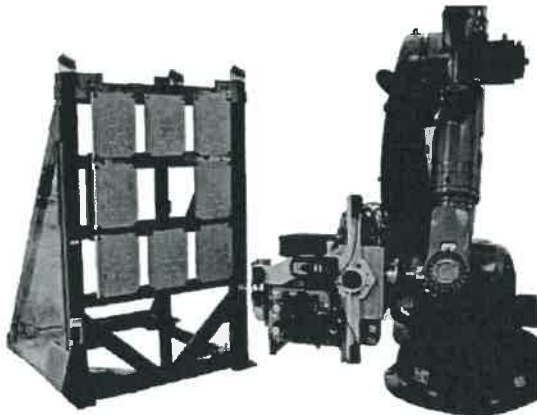


FIGURE 3. Metrology Assisted Robotic Drilling station

CONCEPT IN ACTION

This has been demonstrated in production for over 5 years now, for drilling and riveting in Airbus and in Bombardier along with various installations in research facilities worldwide. The product was initially developed to solve static accuracy processes – drilling, riveting, spotwelding etc but there has been a move recently to look at solutions for dynamic path following applications such as trimming, cutting and finishing. On the face of it, this may seem like a simple extension, streaming in errors to the robot for correction, however, the reality is much more complex. However, this does not mean that nothing can be done to improve path accuracy currently. Nikon Metrology have been working on applying the principles of the ARC solution to increase path accuracy through a combination of specific calibrations and an

automatic point touch-up process called Remember Last Offset.

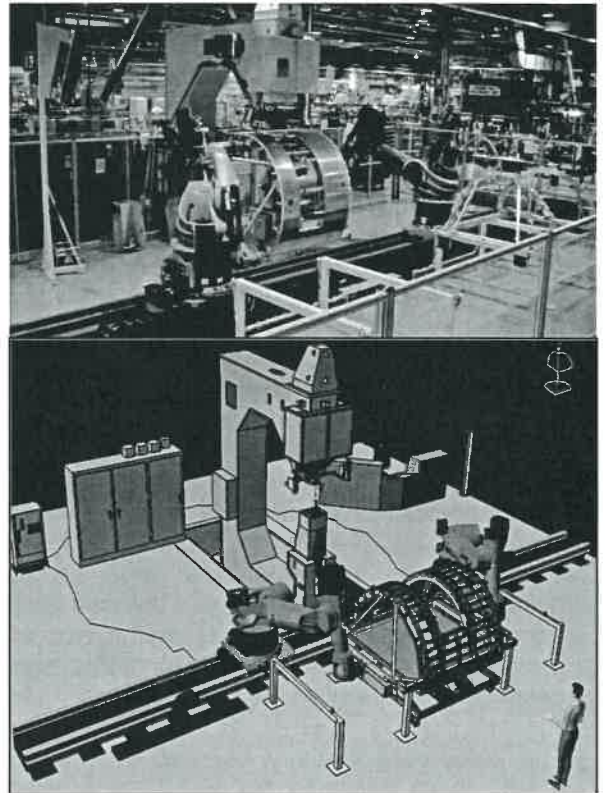


FIGURE 4. Bombardier installation of Adaptive Robot Control

Recent testing has shown that the application of Remember Last Offset can give trimming accuracy of $<0.3\text{mm}$ back to nominal CAD data which is comparable to current processes using a large bespoke machine at a fraction of the cost. This presentation details the methodology and results obtained in that testing as well as learning gained from its development.

DEVELOPMENT OF A MAGNETIC BEARING STAGE WITH REACTION FORCE CANCELLATION

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INTRODUCTION

Magnetic bearing stages have a great potential in the sense that active magnetic bearing enables a stage to have five degrees-of-freedom motion and ultra-low contamination. It comes at the cost of active control systems and relatively weak dynamic stiffness compared to that of fluidic bearings such as hydrostatic oil bearings and air bearings. Magnetic bearing stages are attractive due to its functionality. While their functionality has been proven in many high precision motion control applications [1-3], stages with magnetic bearings seem to put more emphasis on accuracy rather than agility. We introduce our new research to develop a high acceleration magnetic bearing stage. The final goal of our development is to obtain up to 3G (29.4 m/sec²) from a magnetic bearing stage. To cope with high acceleration, our stage has a parallel mechanism to suppress residual vibration. The preliminary design and analysis results will be presented and discussed.

DESIGN OF A MAGNETIC BEARING STAGE

Figure 1 shows an experimental magnetic bearing stage. It is driven by a linear motor with 494 N peak force and its stroke is 280 mm. A fiber optic laser interferometer with a resolution of 20 nm is used for feedback control in the feeding direction. The size of the stage is 283(W)x126(H)x610(L) mm³.

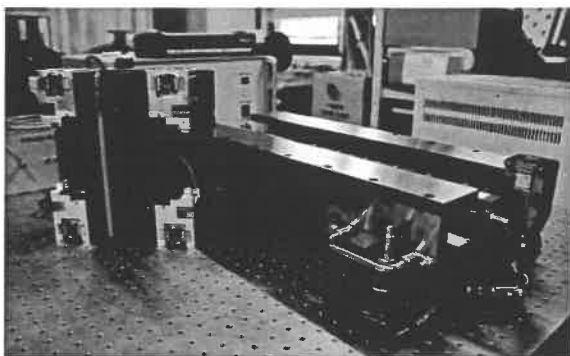


FIGURE 1. Picture of a magnetic bearing stage.

The magnetic bearing actuator designed for the stage is a hybrid type where permanent magnets and electric magnets are combined to make zero bias voltage. Figure 2 shows an electromagnetic actuator module. Each module has three magnetic bearing actuators and four modules are used for the stage, thus, there are eight actuators for the vertical direction and four actuators for the horizontal direction. The maximum load capacity of an electromagnetic actuator is 100 N. The pole area is 600 mm² and the thickness of the permanent magnet is 1 mm and the number of coil turns is 250 for each actuator.

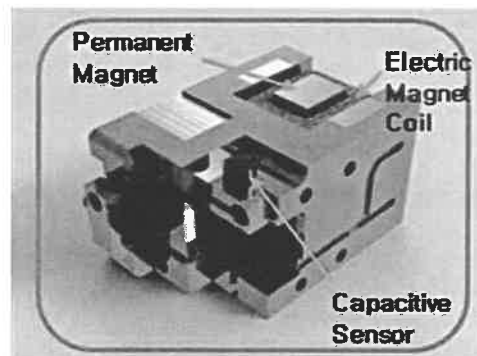


FIGURE 2. Compact design of a magnetic bearing actuator.

The characteristics of the electromagnetic actuator is analyzed using circuit theory and given in Figure 3. As we can see the BH magnetization curve reaches its peak at 0.3 mm gap and the maximum flux density is 0.7 T. Thus, the operational air gap is selected to be 0.3 mm and measured by a capacitive gap sensor. The open-loop stiffness is computed to be -154 N/mm and magnetic force is 150 N at 2 Amps current.

From finite element analysis, important modes appeared in the range of 200 ~ 300 Hz shown in Figure 4. Since these natural frequencies are

relatively high, there seems to be no significant dynamic problems in control system.

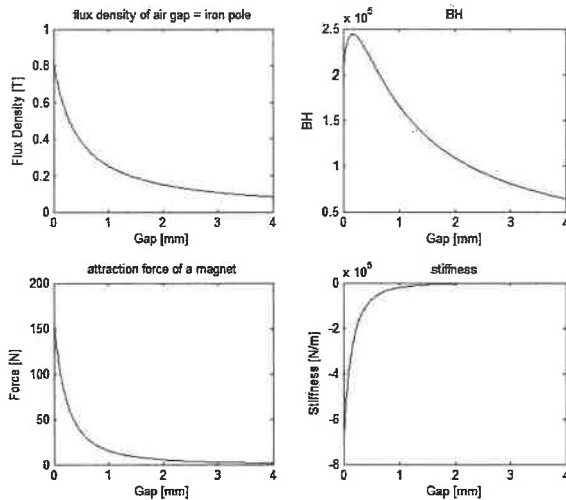


FIGURE 3. Characteristics of the electromagnetic actuator.

Mode	Roll	Trans.-Up/Down	Pitch	Drag-slew
Freq	213 Hz	241 Hz	267 Hz	292 Hz
Shape				

FIGURE 4. Modal analysis result.

REACTION CANCELLATION MECHANISM

The electromagnetic actuators are required to endure high acceleration and will inevitably undergo high residual vibration. To suppress or cancel out the table vibration, a reaction cancellation mechanism is installed next to the magnetic bearing stage as shown in Figure 5. It has a moving mass driven by a cylindrical-type linear motor, which actively generates negative momentum cancelling that from the magnetic bearing stage. The reaction cancellation mechanism is supposed to be synchronized with the stage dynamics, such that the rapid momentum change in the feeding direction of the stage during high acceleration will be absorbed by that of the reaction cancellation mechanism.

In our experimental test with the reaction cancellation mechanism, the input shaping control [4] showed its usefulness in suppressing residual vibration. The input shaper involves real-time shaping or time-delay filtering of the reference command to stable systems with the objective of minimizing the residual vibration. It

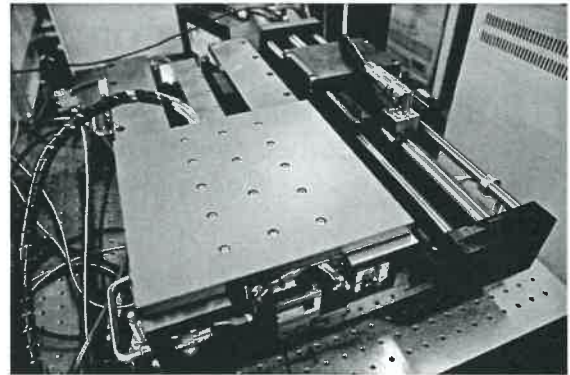


FIGURE 5. A vibration suppression mechanism is installed next to the magnetic bearing stage.

is natural that giving the system an impulse will cause it to vibrate. If a second impulse is given to the system with appropriate amplitude when the system undergoes a half of vibration cycle from the first impulse, it is possible to cancel out the vibration induced by the first impulse with the opposite phase vibration by the second impulse. This is the main idea behind the input shaper. If we have a reasonable estimate of the system's natural frequency, ω , and damping ratio, ζ , then the residual vibration in a second order system resulting from a sequence of impulses can be described by:

$$V(\omega, \zeta) = e^{-\zeta\omega\tau} \sqrt{C(\omega, \zeta)^2 + S(\omega, \zeta)^2} \quad (1)$$

where,

$$C(\omega, \zeta) = \sum_{i=1}^n A_i e^{\zeta\omega t_i} \cos(\omega_d t_i), \quad (2)$$

$$S(\omega, \zeta) = \sum_{i=1}^n A_i e^{\zeta\omega t_i} \sin(\omega_d t_i)$$

In order for Eq. (1) to equal zero, Eq. (2) must both equal zero independently. Therefore, the impulses must satisfy:

$$\begin{aligned} 0 &= A_1 + A_2 e^{\zeta\omega t_2} \cos(\omega_d t_2), \\ 0 &= A_2 e^{\zeta\omega t_2} \sin(\omega_d t_2) \end{aligned} \quad (3)$$

The solution to Eq. (7) in a nontrivial manner is,

$$\begin{bmatrix} A_i \\ t_i \end{bmatrix} = \begin{bmatrix} 1 & K \\ 1+K & 1+K \\ 0 & 0.5T_d \end{bmatrix} \quad (4)$$

where,

$$K = \exp\left(\frac{-\zeta\pi}{\sqrt{1-\zeta^2}}\right), \quad (5)$$

$$T_d = \frac{2\pi}{\omega\sqrt{1-\zeta^2}}$$

If the impulse sequence given at Eq. (4) is convolved with any desired command signal and the convolution product is then used as the command to the system, the convolution product will also cause no vibration. The convolution can easily be implemented as an FIR filter designed from Eq. (4) and (5).

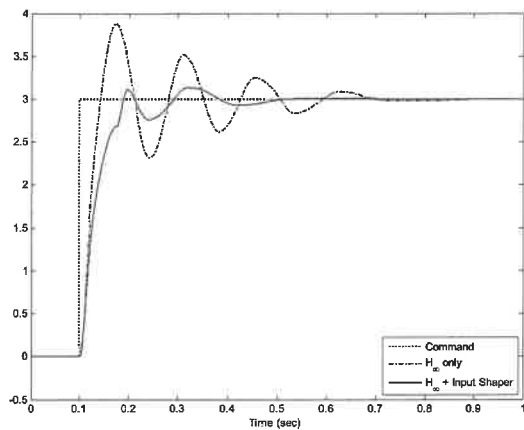


FIGURE 6. Comparison of 3 mm step responses with and without the input shaper.

An H_∞ controller was designed for the reaction cancellation mechanism. When a small step command with a magnitude of less than 1 mm is given to the mechanism, overshoot is hardly seen. When step commands with bigger than 1 mm steps are given, oscillatory behavior could be observed. To cope with the oscillations while radical acceleration, an input shaper was considered being inserted in the inner feedback control loop for the reaction cancellation mechanism. After giving large step commands, the mechanism's damping ratio ζ was estimated to be 0.3035 and natural frequency ω to be 43.96 rad/sec. From Eq. (4) and (5), an input shaper was constructed as an FIR filter and the designed input shaper was put in the servo loop to pre-shape the command before the H_∞ controller. The experimental results with a 3 mm step command are shown in Fig. 6. The input shaper could not cancel out the oscillations completely but, compared to the H_∞ controller only case, it could reduce the oscillations

significantly when the mechanism received large step commands. The convolved command through the input shaper usually increases in length so the overall maneuvering time also increases when an input shaper is used. That is the cost to pay for reducing oscillations.

CONTROL SYSTEM

Because the center of mass of the moving table is not located on the linear motor's moving axis, there are yaw and pitch motions. While 3G acceleration, the forces exerted at each actuator was computed using a dynamic simulation program and its results showed that the vertical actuator needs to endure about 21.1 N and the horizontal actuator about 2.2 N at steady-state. The designed magnetic bearing actuator can endure such amount forces for 3G acceleration, a servo closed-loop controller was designed and implemented in a DS1006 dSpace system. A sampling rate of 25 kHz was used and 6 PID control loops for adopted for 4 vertical levitation control and 2 side positioning control. It was found that a flexible mode from the table appeared at 450 Hz and generated some oscillatory noises. To suppress the flexible mode, a lead-lag controller was also inserted in the servo control loop. Fig. 7 shows the implemented control system for the magnetic bearing stage. The reaction cancellation mechanism was controlled by an H_∞ controller with an additional input shaper and received inverted commands from the magnetic bearing stage controller.

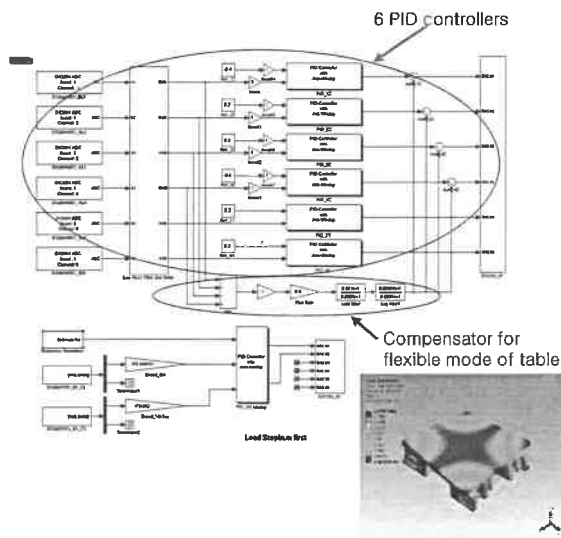


FIGURE 7. Block diagram of the servo control system for the magnetic bearing stage.

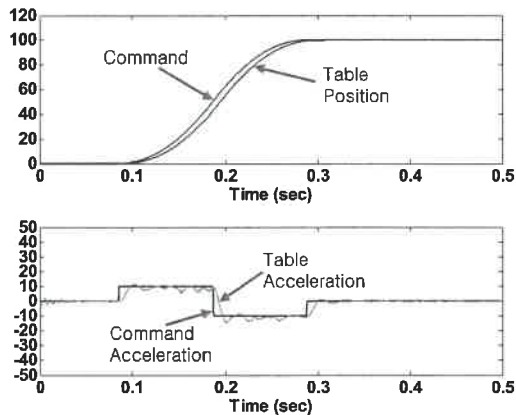


FIGURE 8. Stage response with 1G command.

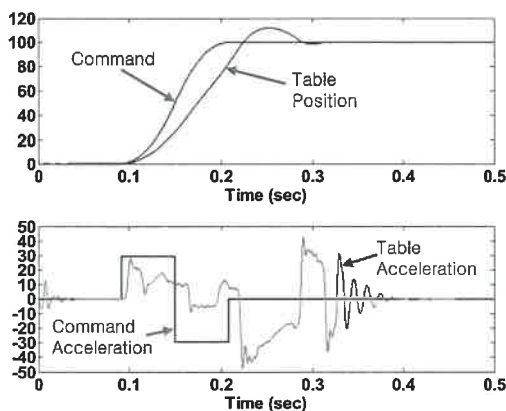


FIGURE 9. Stage response with 3G command.

Fig. 8 and 9 show the experimental results with high acceleration commands. The acceleration was measured by a 4366 accelerometer from Brüel & Kjær. When a command requiring 1G (9.8 m/sec^2) acceleration over 100 mm stroke was injected, the magnetic bearing stage followed the reference command with some lag, but for the 3G (29.4 m/sec^2), the stage barely achieved high acceleration due to a shortage of continuous thrust force from the linear motor. By using feedforward controllers, it may be possible to enhance the initial dynamic responses. The reaction cancellation mechanism successfully cancelled the starting and ending momentum from the magnetic bearing stage and helped reducing the vibrations in the high acceleration stage.

CONCLUSIONS

In this paper we present a prototype design of a high acceleration magnetic bearing stage with a vibration suppression mechanism. Our research aims to achieve up to 3G acceleration using a

magnetic bearing stage and significant residual vibration suppression during acceleration and deceleration time. Using circuit theory and FE analysis, our design was verified for its validity for high acceleration. Dynamic simulation suggested that if the transient response is not controlled with high damping, rubbing may occur between the actuator and the guide way when the acceleration is 3G. Primary experimental results obtained using PID control and an input shaper, were presented in this paper.

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CHARACTERIZATION OF PERIODIC DISTURBANCES IN ROLLING ELEMENT BEARINGS USING AN OPTICAL SENSOR

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INTRODUCTION

There have been considerable improvements in the precision of linear stages that use rolling element bearings due to improved manufacturing techniques. Some applications of linear stages, such as manufacturing and metrology on the sub-micrometer range, require linear motion with sub-micrometer per second precision. For these applications the current level of performance of rolling element bearings are insufficient, which is why magnetic or air bearings are used. Although magnetic and air bearings offer improved performance, due to reduced friction, they are considerably more expensive because of the high tolerances required in their manufacturing. If the precision of rolling element bearings could be improved then linear motion would be less expensive.

In this research we explore the motion of the ball bearings in the race of rolling element bearings that create a fluctuation in the friction force. During constant velocity tracking motion the motion of the ball bearings is periodic, causing the fluctuations to be periodic. The periodic fluctuations in the friction force act like a periodic disturbance on the velocity, causing poor tracking precision.

In this paper the existence of the periodic disturbance and its correlation to the motion of the ball bearings will be demonstrated. This is done by taking measurements of friction force, velocity error, and the rate of the ball bearings during velocity tracking motion. The correlations and spectral densities of the signals are calculated to illustrate the periodic content of each signal and the periodic content that they share with each other. By relating the periodic disturbance to the motion of the ball bearings it may be possible to compensate for the disturbance, thus improving the precision of rolling element bearings.

ROLLING ELEMENT BEARING

In rolling element bearings as the ball bearings transition from the unloaded section of the race to the loaded section of the race, as illustrated in Figure 1,

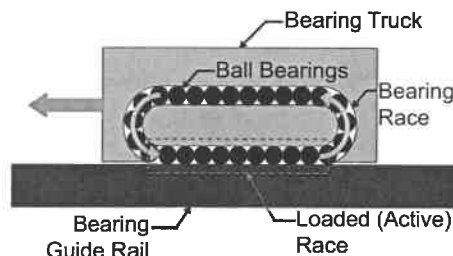


FIGURE 1: Rolling element bearings experience a fluctuation in forces as ball bearings transition from the unloaded to the loaded race.

the bearing experiences a fluctuation in friction force. It is believed that this fluctuation in force causes errors in the tracking velocity. By measuring the rate of the ball bearings traveling through the race and the velocity error it is possible to show their correlation.

During constant velocity tracking the ball bearings move through the race at a constant speed that is half the tracking velocity. This means that the frequency, f , of the ball bearings passing a position in the race is constant:

$$f = \frac{v}{2d}, \quad (1)$$

where v is the tracking velocity, and d is the diameter of the ball bearings. Since it is believed that the fluctuation in friction force occurs every time a ball bearing enters the loaded section of the race, it can be concluded that the frequency of the proposed periodic disturbance acting on the velocity is equal to the frequency of the ball bearings, f . The inverse of f is τ , the period of the ball passage.

As shown in Equation (1) f is proportional to the tracking velocity. This means that τ is inversely proportional to tracking velocity, with a constant of proportionality equal to $2d$. Velocity tracking test were done from 1 mm/s to 20 mm/s and measurements from the optical sensor, which will be discussed later, were used to calculate τ . The ball bearings used in the experiments have a diameter of 4 mm, which means that $\tau = 8/v$. The resulting measured val-

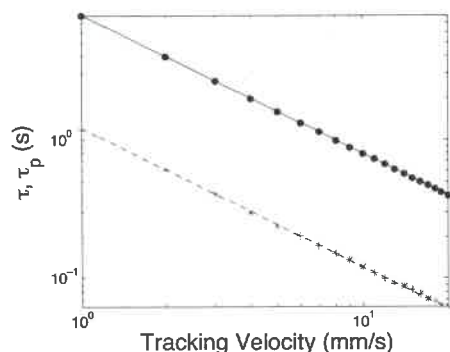


FIGURE 2: Loglog plot of experimentally measured values of τ (●) and τ_p (*) vs. tracking velocity, and fits $\tau = 8.00/v$ (solid) and $\tau_p = 1.22/v$ (dashed) (v in mm/s).

ues for τ can be seen plotted vs. tracking velocity. Also plotted against tracking velocity in Figure 2 is the variable τ_p , which describes the thickness of the peaks in the signal from optical sensor. τ_p is also inversely proportional to velocity with an experimentally determined constant of proportionality of 1.22 mm.

To show a periodic relation at the expected frequency between the fluctuations in velocity and the rate of bearing passage a test setup was built as seen in Figure 3. The bearing truck is held in position with a load cell, which measures the friction force, while the guide rail is moved at a controlled velocity by an air bearing stage. By moving the guide rail instead of the bearing truck we can attach as many sensors to the bearing truck as necessary without creating the challenge of moving the bearing truck along with the sensors. The air bearing stage was chosen to move the rail because it allows for controlled velocity tracking while introducing minimal friction effects.

OPTICAL SENSOR

While trying to observe the ball bearings in the race we would like to ensure that the sensor does not disturb the motion of the ball bearings in the race. Using a small optical fiber to detect balls as they travel through the race is minimally intrusive to the ball bearings motion. Light leaving the tip of the sensing fiber is reflected off a ball bearing as it passes the sensing fiber tip. The reflected light is then measured by the photodetector as illustrated in Figure 4. The red arrows indicate the light from the source while the blue arrows indicate the reflected light. As more light is reflected off the surface of a passing ball bearing the voltage measured by the photodetector increases, indicating the passing of a ball bearing at

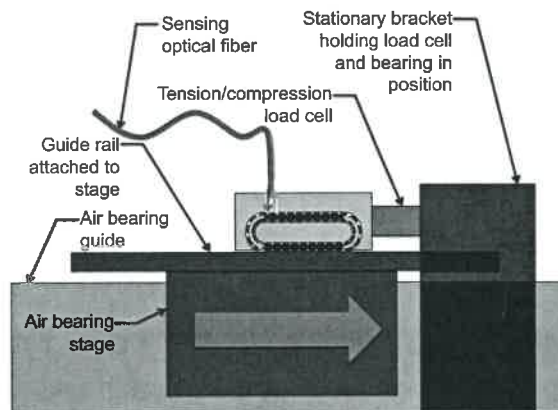


FIGURE 3: Bearing truck is held in position by a load cell while the guide rail is moved by an air bearing stage.

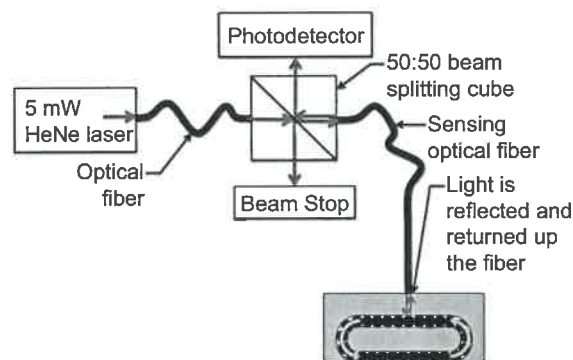


FIGURE 4: Light from the laser is focused into a fiber that is placed inside a bearing truck. The light reflects off the ball bearings, returns up the fiber, and is then measured by the photodetector.

that position in the race.

The sensing optical fiber has a diameter that is 25% of the diameter of the ball bearings. A small hole was machined through the side of the bearing truck to one of the loaded races such that the motion of the ball bearings is not disturbed by the machined hole. The optical fiber is placed within the hole and glued in position so the fiber can capture a detectable amount of reflected light while the tip of the fiber does not make contact with the passing ball bearings. Since the fiber does not contact the ball bearings as they pass and because of the position of the machined hole the sensor is minimally intrusive to the motion of the ball bearings in the race. The resulting signal from the photodetector for a tracking velocity of 10 mm/s can be seen in Figure 5 where each peak indicates a ball passage.

We can calculate that $\tau = 0.8$ s for a 10 mm/s track-

ing velocity test. Figure 5 shows peaks in the photodetector voltage that are spaced approximately 0.8 s apart, as predicted. The auto-spectrum of the photodetector signal, which is shown in Figure 6, is calculated and its plot can be seen in Figure 6. From Equation (1) we would expect to see increases in the auto-spectrum at $f = 1.25$ Hz. Vertical red-dashed lines are used to illustrate the fundamental frequency $f = 1.25$ Hz and higher harmonics. Figure 6 shows that Equation (1) is indeed a good approximation for the frequency of the ball bearings moving through the race for tracking velocity motion.

The signal from the photodetector can be modeled as a Gaussian pulse train:

$$s(t) = \sum_n A \exp \left[- \left(\frac{t - n\tau - t_0}{\tau_p} \right)^2 \right], \quad (2)$$

where A is the amplitude of a pulse, τ_p is the width of a pulse, and t_0 is the time of the zeroth pulse. Like τ , τ_p is inversely proportional to the tracking velocity; the experimentally determined relationship is $\tau_p = 1.22/v$ for v in units of mm/s (see Figure 2).

RESULTS AND DISCUSSION

The friction force, velocity error, and photodetector voltage were recorded for a tracking velocity of 10 mm/s, corresponding to $f = 1.25$ Hz and $\tau = 0.8$ s, using the test setup depicted in Figure 3. To illustrate the relationship between periodic fluctuations in the friction force causing fluctuations in the tracking velocity, the auto-correlations are shown in Figures 7 and 8. The signals were low-pass filtered and detrended so that the low frequency components are more prevalent in the auto-correlations and the spectral densities, which will be discussed later.

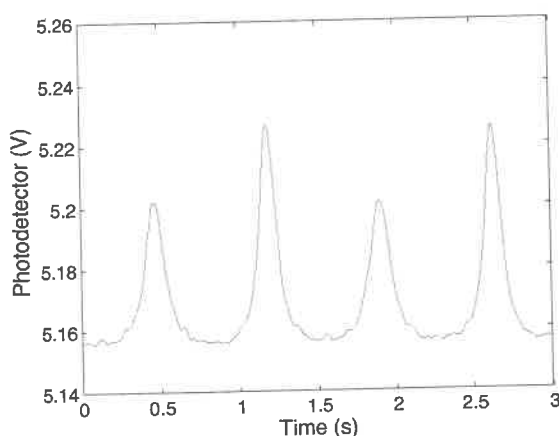


FIGURE 5: Signal from photodetector for a tracking velocity test at 10 mm/s (corresponding to $\tau = 0.8$ s).

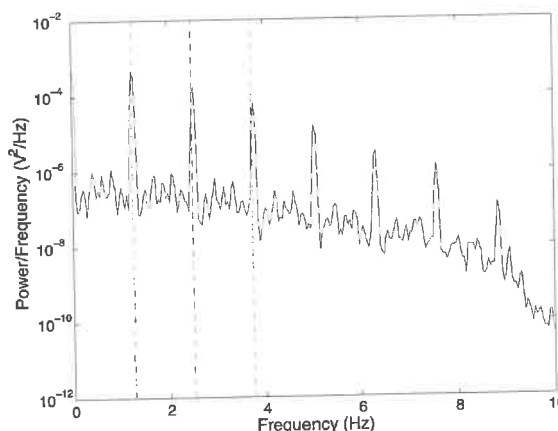


FIGURE 6: Auto-spectrum density calculated from the photodetector signal for a tracking velocity test at 10 mm/s (corresponding to $f = 1.25$ Hz).

Correlation times where the auto-correlation of a signal increases means there is periodic content within the signal whose period is equal to the correlation time at which the auto-correlation increases. As seen in Figures 7 and 8 there are increases in the auto-correlations at a correlation time of 0.8 s, indicating that there is some periodic content present with a period of 0.8 s.

To determine how the periodic content of the friction force and velocity error are related to the periodicity of the ball bearings their cross-spectra were calculated and zoom processing was used to help resolve the cross-spectra in the range of the frequency estimated by Equation (1) [1]. The cross-spectrum between the friction force and the photodetector voltage is shown in Figure 9 while the cross-spectrum between the velocity error and the photodetector voltage is shown in Figure 10. The vertical red-dashed lines in the plots of the cross-spectra are used to indicate the fundamental frequency $f = 1.25$ Hz and higher harmonics. Increases in the cross-spectrum of two signals at some frequency indicates that the two signals share some periodic content at that frequency.

SUMMARY

The plots of the auto-correlations of the friction force and the velocity error, shown in Figures 7 and 8 respectively, indicate that there is some periodic content present in the friction force and velocity error whose period is equal to the period of the ball bearing motion predicted by τ . While the auto-correlations show the presence of the periodic content the cross-spectras, shown in Figures 9 and 10, show that the friction force and the velocity error

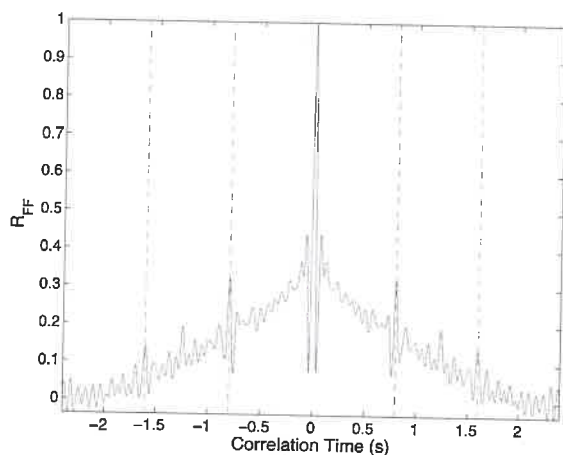


FIGURE 7: Auto-correlation of friction force for a tracking velocity test at 10 mm/s (corresponding to $\tau = 0.8$ s).

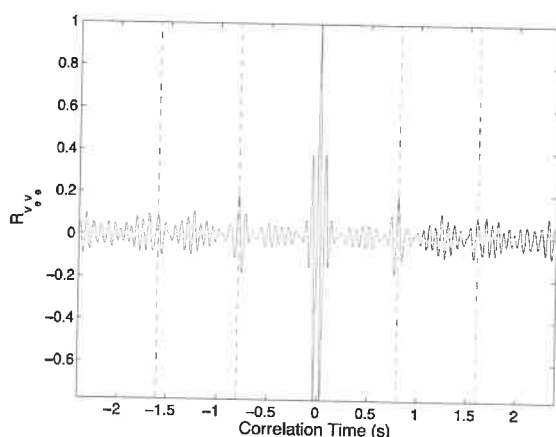


FIGURE 8: Auto-correlation of velocity error for a tracking velocity test at 10 mm/s (corresponding to $\tau = 0.8$ s).

share periodic content with the optical sensor at the frequency predicted by Equation (1). This indicates that there may be some correlation between the periodic disturbance in the velocity that is caused by the fluctuating friction force and the motion of the ball bearings.

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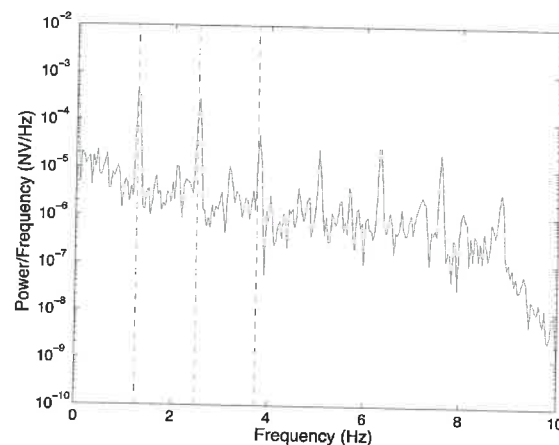


FIGURE 9: Cross-spectrum of friction force and photodetector voltage for a tracking velocity test at 10 mm/s (corresponding to $f = 1.25$ Hz).

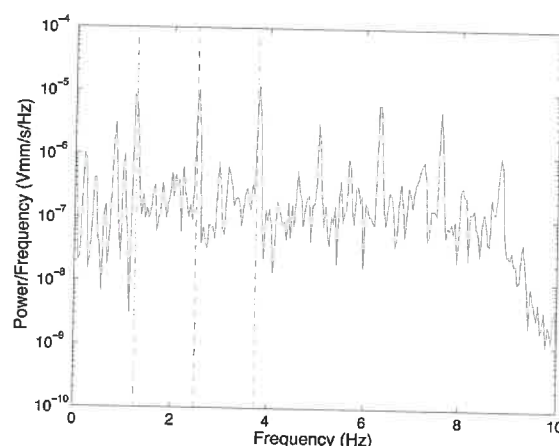


FIGURE 10: Cross-spectrum of velocity error and photodetector voltage for a tracking velocity test at 10 mm/s (corresponding to $f = 1.25$ Hz).

VIBRATION SUPPRESSION FOR IMPULSIVE MOTION IN DIRECT DRIVE LINEAR STAGE

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ABSTRACT

This paper presents the results of a study on the impulse decoupling technique applied to a fast linear direct drive stage. A spring-damper system is implemented between the linear motor and the supporting structure. Analyses are carried out to find the effective stiffness and damping coefficients of the system for achieving low force transmissibility and jerk. The technique is applied to a stage capable of reaching the acceleration of 300 m/s^2 in a 500 mm travel range. It is experimentally verified that for a harmonic impulse input the jerk can be reduced from 100 m/s^3 to 0.4 m/s^3 with the implementation of the decoupling strategy. The analytical model developed and verified experimentally can serve as a useful design tool for incorporating the impulse decoupling technique in fast linear motion stages.

INTRODUCTION

Linear direct drive motor is increasingly being used in high performance equipment and machine tools. This is mainly due to its capability to achieve high acceleration motion which is critical for high throughput operations. However, the impulsive high reaction force generated by theforcer coil is directly transmitted to the machine base and the various frequency modes are excited. The resulting oscillation of the machine structure will compromise the precision requirement and the dynamic properties of the entire machine. There are several methods to deal with the vibration effects. One of them is active vibration control where an actuator applies oscillatory force to compensate the vibration based on the sensing signal. Such method is proven effective in reducing unwanted vibration for a wide range of frequency. Another method is known as jerk minimizing technique. In this approach, acceleration profile is shaped to prevent sharp corners that can induce infinite

jerk, which is the rate of acceleration change. In one example [1] a well designed displacement motion curve can reduce the residue vibration by 70% as compared to the vibration resulted from the conventional S-curves. However for impulsive motion with extremely short rise time, the above methods will not be effective. Previous work suggested that the impulse decoupling strategy [2][3] will improve the machine dynamics. Our study correlates the properties of spring-damper system that links the motor and the machine structure to the force transmissibility and jerk profile. It begins with the analysis of the entire stage system model. The structural behaviors with and without the spring-damper system are compared. The simulated results are verified experimentally.

SYSTEM MODELLING

A mechanical linear direct drive mounted on a machine structure through a spring-damper system is shown in FIGURE1. In this model, masses of the carriage, linear motor (specifically its stator) and machine structure/base are represented respectively by m_1 , m_2 , and m_3 . Spring constant and damper coefficient between carriage and the motor stator are respectively k_1 and c_1 . Spring constant and damper coefficient between stator and the machine base are respectively k_2 and c_2 .

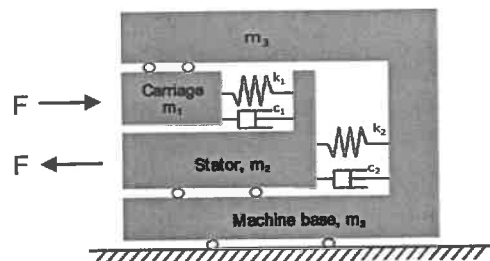


FIGURE1. Model of linear motor stage on machine base with decoupled stator.

Assume the input force F acts towards the right of m_1 and the accelerations of m_1 , m_2 , and m_3 are positive towards the left. The notations x_i , $\dot{x}_i$, $\ddot{x}_i$ represent respectively the displacement, velocity and acceleration of m_i where $i = 1, 2, 3$. Equations of motion can be expressed as follows:

$$\begin{aligned} m_1 \ddot{x}_1 + c_1 \dot{x}_1 - c_1 \dot{x}_2 + k_1 x_1 - k_1 x_2 &= F \\ m_2 \ddot{x}_2 - c_1 \dot{x}_1 + (c_1 + c_2) \dot{x}_2 - c_2 \dot{x}_3 - k_1 x_1 + (k_1 + k_2) x_2 - k_2 x_3 &= -F \\ m_3 \ddot{x}_3 - c_2 \dot{x}_2 + c_2 \dot{x}_3 - k_2 x_2 + k_2 x_3 &= 0 \end{aligned}$$

Expressing the equations in the state space form, gives

$$\dot{a} = Aa + Bu \quad (1)$$

where $a = [x_1 \ \dot{x}_1 \ x_2 \ \dot{x}_2 \ x_3 \ \dot{x}_3]^T$, u is the input value which is 1 and

$$A = \begin{bmatrix} 0 & 1 & 0 & 0 & 0 & 0 \\ -k_1 & -c_1 & k_1 & c_1 & 0 & 0 \\ m_1 & m_1 & m_1 & m_1 & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 & 0 \\ k_1 & c_1 & -(k_1 + k_2) & -(c_1 + c_2) & k_2 & c_2 \\ m_2 & m_2 & m_2 & m_2 & m_2 & m_2 \\ 0 & 0 & 0 & 0 & 0 & 1 \\ 0 & 0 & k_2 & c_2 & -k_2 & -c_2 \\ 0 & 0 & m_3 & m_3 & m_3 & m_3 \end{bmatrix}$$

$$\text{and } B = [0 \ F/m_1 \ 0 \ -F/m_2 \ 0 \ 0]^T.$$

Using Equation (1), the displacement, velocity, acceleration and jerk of the masses, and the force transmitted to the machine base can be obtained.

ANALYSES

To investigate the impact force and jerk, the masses of the model are set as follows: $m_1 = 1$ kg, $m_2 = 42$ kg, and $m_3 = 134$ kg. The input force acting on m_1 is a periodic ($j = 0, 1, 2 \dots$) with a period of 0.2 s as follows:

$$F = \begin{cases} -300 \sin(20\pi t), & 0.2j \text{ s} < t \leq 0.1(2j+1)\text{s} \\ 300 \sin(20\pi t), & 0.1(2j+1)\text{s} < t \leq 0.2(j+1)\text{s} \end{cases} \quad (2)$$

The force and displacement profiles are plotted respectively in FIGURE 2 and FIGURE 3. For $m_1 = 1$ kg, the maximum acceleration is 300 m/s² and its profile will have discontinuities for such input profile.

For the case that the motor stator m_2 is fixed on the machine base, which is a common practice of mounting the motor in machine tools and semiconductor equipment, such harmonic motion input profile will lead to jerk in the machine base as high as 100 m/s³ for the input

force of Equation (2) (See FIGURE 4). If we introduce an isolator, in the form of damper-spring system between the linear motor stator and the machine base, the force transmitted to the machine is expected to be smaller.

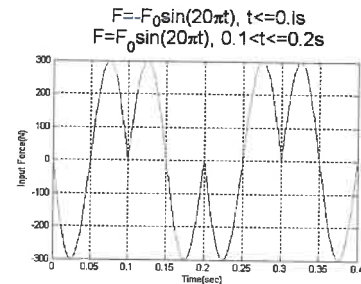


FIGURE 2. Force F of Equation (2) applied to carriage m_1 for cycle time of 0.2 s.

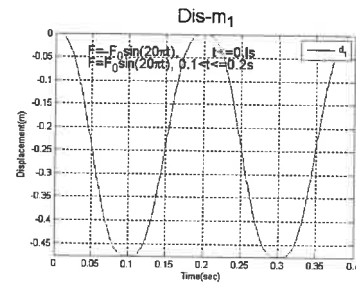


FIGURE 3. Displacement of carriage m_1 as a result of force F of Equation (2) applied to carriage m_1 for cycle time of 0.2 s.

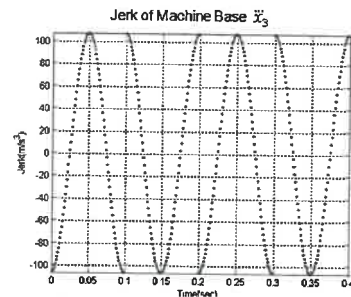


FIGURE 4. The jerk of the machine base with the force of Equation (2) applied to the carriage m_1 . It shows that the jerk without decoupling can reach 100 m/s³.

Force transmitted to machine base can be expressed as $F_T = k_2 x_2 + c_2 \dot{x}_2$, where k_2 and c_2 are respectively the spring constant and damping coefficient of the isolator. The transmissibility ratio (TR) is given by

$$TR = \frac{F_T}{F} = \left\{ \frac{1 + (2\zeta r)^2}{(1 - r^2)^2 + (2\zeta r)^2} \right\}^{\frac{1}{2}} \quad (3)$$

where $r = \omega/\omega_n$ is frequency ratio, $\zeta = \frac{c_2}{2\sqrt{k_2 m_2}}$ is the damping ratio, $\omega_n = \sqrt{\frac{k_2}{m_2}}$ is the natural frequency of m_2 , and ω is the forcing frequency. Using Equation (3), it can be deduced that when $r \leq \sqrt{2}$, TR is less than 1, and increasing r or decreasing ζ will reduce the TR. FIGURE 5 is the TR expressed in k_2 and c_2 . Within the box, the TR is smaller than 0.5. The maximum values of k_2 and c_2 for the TR smaller than 0.5 are respectively 10 kN/m and 500 N·s/m.

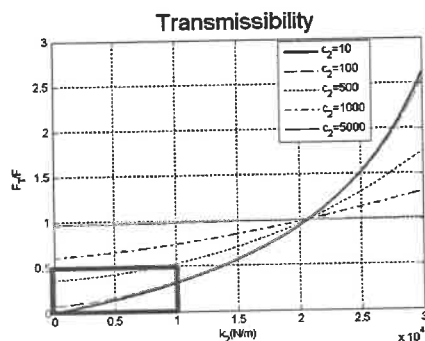


FIGURE 5. Transmissibility ratio F_T/F in terms of the decoupling stiffness k_2 and damping coeff c_2 .

When $k_2 = 1$ N/m and $c_2 = 100$ N·s/m, the jerk on the machine base is less than 0.4 m/s³. FIGURE 6 shows the profile of the jerk and FIGURE 7 the force transmission profile when the spring-damper system is implemented. The decoupled linear motor stator m_2 moves with the displacement profile shown in FIGURE 8 and the machine base m_3 will oscillate as shown in FIGURE 9 if m_1 continues with the cyclic motion.

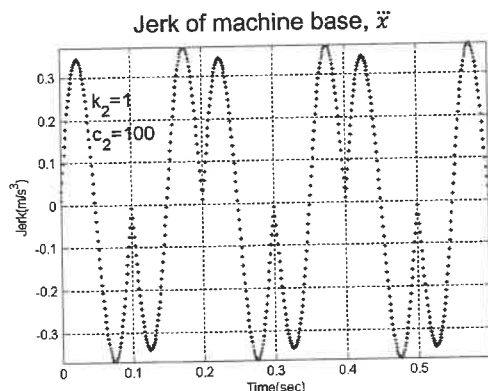


FIGURE 6. Jerk in machine base when it is decoupled from the linear motor with $k_2 = 1$ N/m, $c_2 = 100$ N·s/m.

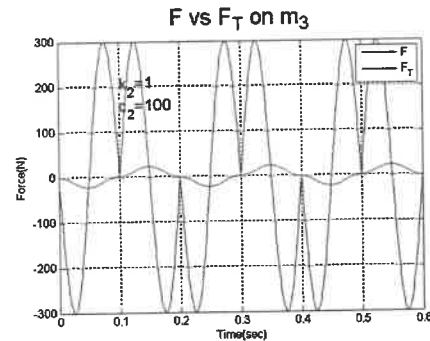


FIGURE 7. Input force F vs force transmitted to the machine base when $k_2 = 1$ N/m, $c_2 = 100$ N·s/m.

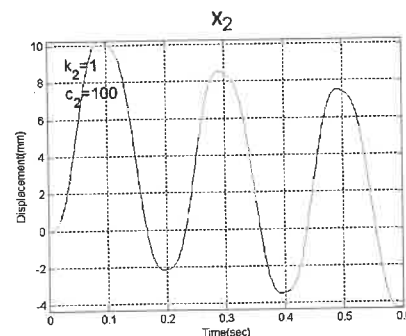


FIGURE 8. Displacement of linear motor stator when $k_2 = 1$ N/m, $c_2 = 100$ N·s/m over first 3 cycles.

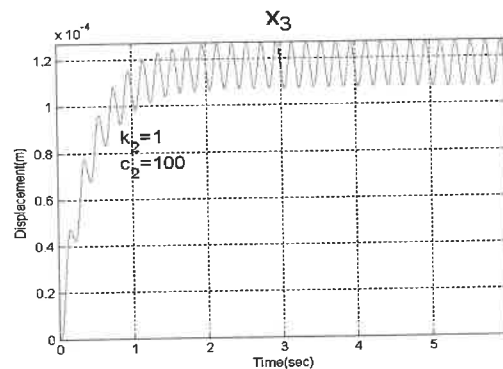


FIGURE 9. Displacement of the machine base when $k_2 = 1$ N/m, $c_2 = 100$ N·s/m for a prolonged period of time.

EXPERIMENT

The numerical results are verified using the experimental setup shown in FIGURE 10. In the setup, the carriage m_1 guided by the primary guideway is moved by a forcer coil which is sandwiched in a u-channel type permanent magnet stator. The stator m_2 is in term mounted on a slider, namely secondary guideway, which

is linked to the machine base m_3 through a spring-damper system. The displacement profile of the stator is measured as the carriage is moved with the input force as expressed in Equation (2).

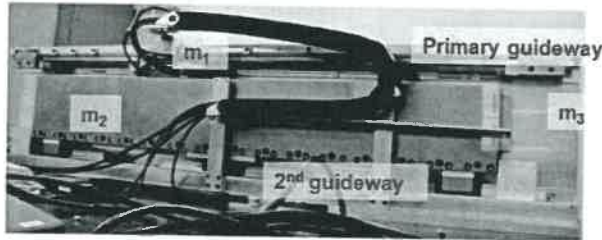


FIGURE 10. Photo of a direct drive linear motor with its stator movable along a secondary guideway and linked to the machine base through a spring-damper system (not shown).

In the experiment, the carriage m_1 moves from end to end of the stage. At each end position, the displacement of the stator m_2 is measured and the values are shown in FIGURE 11 for 6 stops. It can be seen that the stator m_2 almost displaces the same distances for both back and forth motion for the first few cycles. The centre position of the m_2 however has the trend of shifting towards one side of the machine base. It is expected that as such motion cycle continues, the centre position of the m_2 will reach a stationary position where the system will be at a steady state as suggested by the displacement profiles of m_3 in FIGURE 9. On the other hand, when the linear motor stator is fixed on the machine base, which is the common practice for mounting the motor in the machine, the displacements of the machine base, are quite large as shown in FIGURE 12. This is a result of the high jerk shown in FIGURE 4.

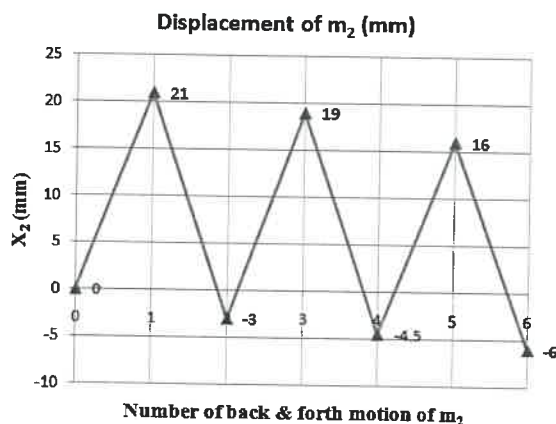


FIGURE 11. Measured displacement of the stator when $k_2 = 1 \text{ N/m}$, $c_2 = 100 \text{ N-s/m}$.

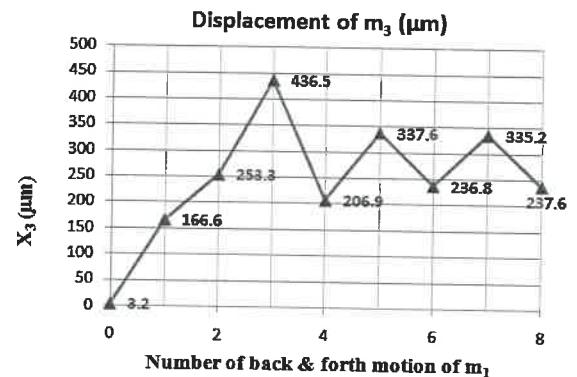


FIGURE 12. Measured displacement of the machine base with the stator fixed on it.

The experimental results agree with the theoretical results in term of motion trends of the decoupled linear motor stator. The large discrepancies in their values are mostly due to the discontinued motion of the carriage and inaccurate damping coefficient and the stiffness used in the experiment.

CONCLUSION

In this study, an analytical model has been developed for the impulse decoupling strategy for the linear motor stage. For the decoupling system of spring stiffness of about 1 N/m and a damping coefficient of about 100 N-s/m implemented for the particular masses in the stage, the force transmitted to the machine base is reduced by 90%, and the jerk of machine base is reduced from 100 m/s^3 to 0.4 m/s^3 . The analytical model verified experimentally can serve as a useful design tool for fast linear motion stage incorporated with the impulse decoupling technique.

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CONDITIONS AND LIMITATIONS OF THE LOW-BANDWIDTH VISUAL-SERVO LOOP IN MICRO-POSITIONING APPLICATIONS

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INTRODUCTION

The present research studies the dynamics of the *single-camera (single-CAM) single-image processing thread* visual servo loop in micro-positioning applications. The study is conducted on a previously introduced control system utilized for the positioning of an XY table. Analysis and experimental results expose fundamental limitations in low-bandwidth vision-based servo-mechanisms when applied to high-resolution position control. Although sensor resolutions on the order of $2.5\text{ }\mu\text{m}$ are reliably achieved through adequate image processing, the overall best achievable resolution for in-plane closed-loop positioning of the stage is found to be on the order of 100's of micrometers. This aspect is attributed to errors in the approximation of the feedback signal; such approximation is required to compensate for intermittencies and delays associated with the slow dynamics of the imaging sensor and the image processing thread. Though this approximation would certainly be considered acceptable for macro-positioning applications, micro-positioning systems might require additional elements to aid in achieving the desired resolution levels. The conditions for achieving smooth output motion in vision-based control systems when integrated in manufacturing equipment are also presented.

EXPERIMENTAL SETUP

Implementation is carried out on an XY-stage as shown in Figure 1. The desired displacement command of the stage in a planar environment is emulated, in one end of the kinematic chain, by a cross-hairs target on a liquid-crystal display (LCD) [2]. On the other end of the kinematic chain a camera observes the active target and provides visual feedback information utilized for position control of the tool. This system was first introduced in [1], and further investigated in [2].

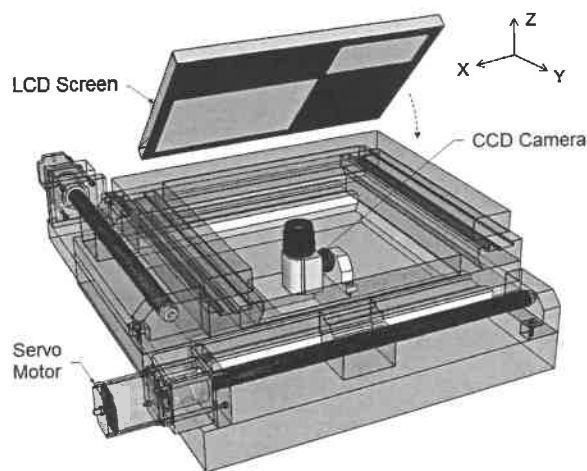


FIGURE 1. 3D CAD model of XY-stage.

The analysis presented next concerns a *single CAM- single image processing thread*-configuration, where a single camera and a single graphic processing unit (GPU) are used to acquire and process the target images.

THE HYBRID COMMAND ISSUING PROTOCOL

In the hybrid command generation protocol, commands equal to a multiple of one LCD pixel are issued by moving the target on the LCD by that specific distance, and setting the reference on the image plane to the origin (i.e. the camera's principal point (PP)). In this case the control system operates as a regular tracking system. For commands smaller than one LCD pixel or not equal to an integer multiple of pixels, the target is displaced by the smallest integer number of pixels that is larger than the desired commanded displacement and the reference signal is set to a value different than PP; the value of the reference is less than 1 LCD pixel size. Figure 2 shows the proposed control structure, where the path plan is generated off-line, block C represents the controller, G

represents the dynamics of a single axis, $\hat{G}$ represents the mathematical model of a single axis, V represents the camera- and image processing thread-dynamics and $\hat{V}$ its corresponding model. Variable x_i^{Target} is the displacement of the target referenced to the LCD frame. The reference signal to the control system is defined as $x^{ref} = x^{Target} - x^{fine}$. The value x^{fine} is the desired displacement command, where $|x^{fine}| < 1$ LCD pixel. Proof of convergence of x (position of the stage) to x^{fine} when setting $x^{ref} = x^{Target} - x^{fine}$, as well as other considerations for implementation of this method, are provided in [3].

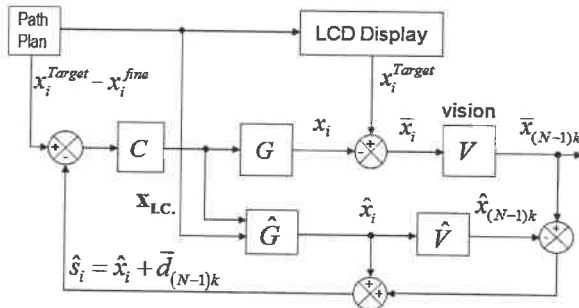


FIGURE 2. Model-aided control system scheme represented in the discrete domain with discrete sampling variable i .

In the proposed hybrid protocol the states of the model $\hat{G}$ are enforced to follow the relative stage position, $\bar{x}_i$, by updating the initial conditions of the model states, $\mathbf{x}_{LC}$, every time the position of the target, x_i^{Target} , is displaced on the LCD. This procedure requires the synchronization of the different experimental components, later called processing threads. The conditions for adequate operation of the system in Figure 2 are presented next. Additionally, these conditions are extended to more common visual tracking architectures for position control of NC machines.

CONDITIONS OF OPERATION

Dynamic Target Tracking

The update of the model initial conditions should only occur right after the position of the cross-hairs has changed with respect to the LCD frame. Hence, adequate command issuing can only be achieved if the host thread, in charge of generating and displaying the dynamic target,

and the control thread, where the model-based closed-loop system is run, are synchronized (Figure 3).

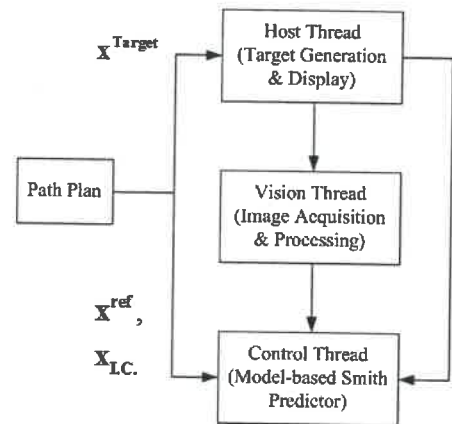


FIGURE 3. Multi-thread communication diagram.

Once a new target imaged has been displayed, the host thread must notify the control thread so that the latter updates both the initial conditions of the model and the reference signal. The time diagram in Figure 4 shows how the three processing threads operate in a synchronized manner. Communication times between threads are considered to be small, with respect to other processing times, and are assumed to be negligible in the analysis. At t_0 the host thread displays a new target image, x^{Target} . Once the image has been updated on the LCD, the vision thread starts the acquisition process at t_0^D , and $\mathbf{x}_{LC}$ and x^{ref} are updated at t_0^{update} in the control thread (with $t_0^D = t_0^{update}$). At this point the stage starts moving according to the initial conditions of the model and the control effort tries to bring the model states to the reference position. If the resulting velocity from the control effort is too high the tracking error will be reduced to zero rapidly. In Figure 4 it takes the control system Δt_0^{fine} seconds to bring the tracking error to zero, according to the model feedback signal. The stage then waits motionless for Δt_0^{wait} seconds until a new target image is displayed at t_1 . This wait time is actually a drawback as it introduces discontinuities in the axis output velocity.

position of the stage obtained from the camera is available only every k samples (where k is a constant), and provides information about the plant k samples late. Then for instance, at $i = k$ the estimated feedback signal, $\hat{S}$ in Figure 2, is $\hat{S}_k = \hat{x}_k + (\bar{x}_0 - \hat{x}_0)$. The approximate correcting factor $(\bar{x}_0 - \hat{x}_0)$ can be applied during the entire interval $i \in [k, 2k)$ until a new feedback point is generated by the vision block at $i = 2k$. Let the correcting factor be defined in the discrete domain as $\bar{d}_i = \bar{x}_i - \hat{x}_i$. Then in general $\hat{S}_i = \hat{x}_i + \bar{d}_{(N-1)k}$, where $i \in [Nk, (N+1)k)$ and $N \in \mathbb{N}_0$. The error between the ideal and actual feedback signals is defined as

$$\begin{aligned}\bar{e}_i &= s_i - \hat{S}_i \\ \Rightarrow \bar{e}_i &= \bar{x}_i - (\hat{x}_i + \bar{d}_{(N-1)k}) \\ \Rightarrow \bar{e}_i &= \bar{d}_i - \bar{d}_{(N-1)k}\end{aligned}\quad (2)$$

Discrepancies between the ideal and actual feedback signals in model-aided vision-based control, given by (4), are responsible for inaccuracies in the output displacement of the tool. It is important to stress that even if the vision block was capable of operating at the controller's frequency ($k=1$), i.e. without intermittencies, the error between the ideal and actual feedback signals would still be different from zero due to the processing delay. In such case, $\bar{e}_i = \bar{d}_i - \bar{d}_{(N-1)}$ for $i \in [N, N+1)$ and $N \in \mathbb{N}_0$. Error $\bar{e}_i$ would only be zero if no intermittencies and no delays were present in the vision block feedback signal; in such case, however, no modeling would be required. This is a fundamental limitation associated with delays and intermittencies in *single CAM- single processing thread*-visual servoing, when utilized in high-resolution positioning applications.

EXPERIMENTAL RESULTS

A basic hysteresis-path plan for fine in-plane displacement is generated (Figure 6, top). Figure 6, top, shows the actual stage output motion plotted on top of the desired in-plane path. Figure 6, bottom, shows the desired constant velocity pattern for X-axis motion when enforcing the frequency and velocity conditions.

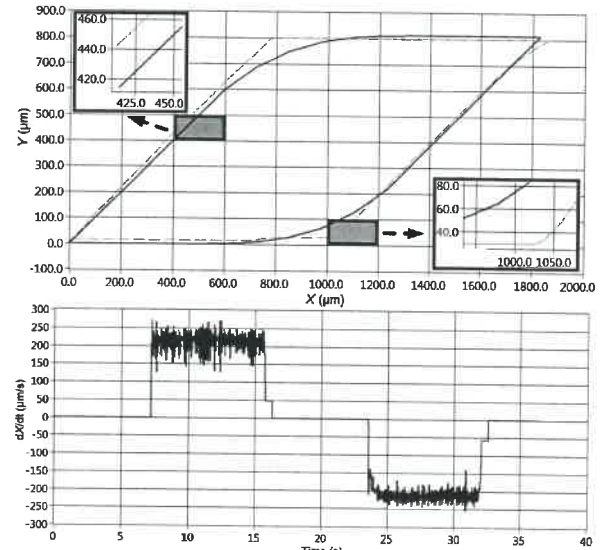


FIGURE 6. Experimental results: (top) Desired (cont.) and experimental (dashed) in-plane motion; (bottom) Output velocity, X-axis.

CONCLUSION

Overall best achievable resolution for in-plane positioning is on the order of 100 μm . For the *single CAM- single image processing thread*-configuration, accuracy errors are associated with the feedback signal, which is approximated through modeling and old sensed data.

ACKNOWLEDGEMENTS

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HYDRAULIC ACTUATOR FOR POWER JOINT MECHANISM

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INSTRUCTIONS

In the developed countries some fields of the working process in ship building industry are still using labor of worker because they are not developed in the automatic processes and they have very bad working conditions. Even though the industries of ship building have been tried to develop the automatic process, they did not obtain the remarkable results and they are also still trying to develop the automatic process in ship building. In the developed countries most of the working process of the automobile manufacturing company is performed in almost perfectly automatic process. And the working materials and equipments manufactured by major large companies are used in most of automobile manufacturing process because they have very good quality for working process.

In many companies of ship building industry large parts of the working process are still dependent on labor of operators. And also in the other developed countries they are still dependent on it. Though some of the companies are tried to develop the automatic process, they did not obtain the considerable results. The real site of working process in ship-building industry has the difficulty to be changed into the automatic working process. It has the bad environment and high pressure of working load for worker to work in long time. The working worker carrying several heavy tools should be stayed to do given job in closed working space or inside internal wall of ship block and in long time. The scope of working motion is expressed in angle of working tool motion to move in required conditions with carrying a variety of tools such as spray gun, grinder and welder. As this scope becomes larger and it is possible to do the handling work in complex shape and narrow block space. The driving axis of working motion is the number of axis of link type to construct the main axis of working assistant system. Generally, the working motion becomes much smoother as the number of this axis becomes larger. The kind of working tools is

generally classified into several categories to do the handling work in different handling motion by using different handling tools such as spray gun, grinder and welder.

The number of driving axis of related motion is defined in relation with arm and wrist motion of worker when the operator in working site is doing the handling work. The working velocity is defined as the velocity of handling tools to move in working path and it is closely related with the productivity of handling work. The working load is defined as the total force to be pulled and pushed at end of the working assistant system or to be hanged on it [1, 2, 3].

HYDRAULIC ACTUATOR FOR POWER JOINT MECHANISM

The working assistant system is designed to be used without any overload to body joints when the high vibration of drilling tool is occurred. The reaction of drilling pressure and self-weight of working tool are transferred to the shoulder and arm of heavy worker and the accumulated fatigue in body system is occurred. As results, the productivity of handling work is degraded and it is designed to prevent the industrial accident from doing the handling work. The working assistant system is composed in several complex mechanism of multi- axis link type to do handling work in a variety of degree of freedom and it is possible to be wearable and to amplify the articulation power of worker[4,5,6]. To do this, the joints of working assistant system are rotated by electrical motor or hydraulic actuator. The hydraulic actuator is installed on joint link between thigh and shin to perform powerful linear motion. This actuator takes charge of the roll to rotate shin link. Like this, if the link is rotated by the actuator to do linear motion the joint has the disadvantage to be interfered by the actuators. In hydraulic system there are several kinds of hydraulic actuator to make rotating motion. This kind of actuator has not only low efficiency of power but also the difficulty of exact closing motion because of pressure

difference and flow leakage between high pressure room and low pressure room in hydraulic actuator system. Therefore, the new hydraulic cylinder motor is proposed to solve demerits of hydraulic motor and to make rotating motion possible. Figure 1 shows the principle of operation of proposed system that is expressed in counterclockwise direction of follower pulley. The hydraulic actuator is the apparatus to convert linear motion into rotating motion by combination of one or more than two pulleys, wire-rope and directional change valve.

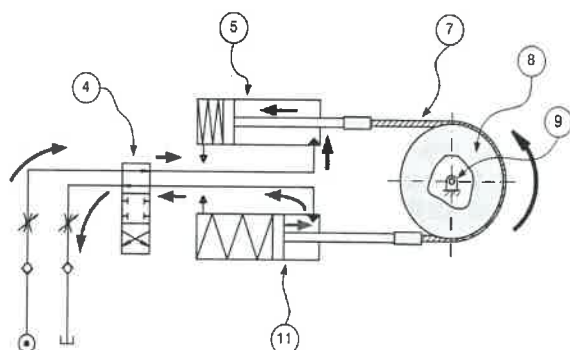


Figure 1. Operating mechanism of hydraulic cylinder and follower pulley in counter clockwise direction

The special feature of this system is to realize more than 1/2 rotation motion of idle pulley to be installed on link-joint parts. And also the special feature of this system has the simple structure and high efficiency of power because of small leakage of operating oil compared with hydraulic motor to use flow power directly. And it has the possibility of exact closing motion because of adjustment of back pressure in internal pressure room and rotation motion in less than or more than half rotation. Figure 2 shows the internal mechanism of hydraulic actuator that has follower pulley and wire rope. This hydraulic actuator is designed in different size as amount of required joint power. The treatment of hydraulic supply line is considered in design of compact shape of joint link. Figure 3 shows hydraulic actuator installed on joint link in working assistant system. The wrist joint has 3 DOF mechanism that the additional 1 DOF mechanism is added to mechanism of 2 DOF lower arm part. The shoulder joint has 3 DOF mechanism that the additional 1 DOF mechanism is added to mechanism of 2 DOF upper arm part.

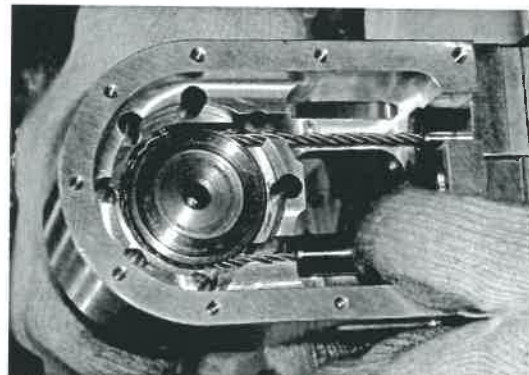


Figure 2. Mechanism of hydraulic actuator

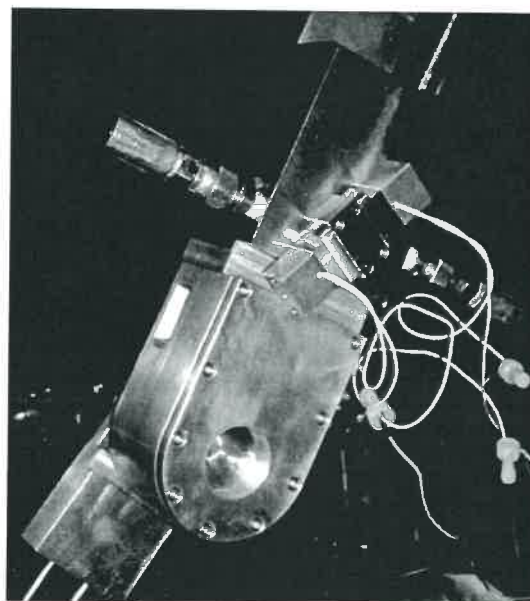


Figure 3. Hydraulic actuator installed on joint

The experimental models of actuator, joint and link are manufactured and tested as shown in Figure 3. The power pack supplies hydraulic power to actuator. The hydraulic line, electrical line and valves to control the system motion are shown in Figure 3. Figure 4 shows the experimental test equipment of solenoid valve driver. After setting the gap between solenoid valve spool and coil the steady working of solenoid valve is tested by Analog to PWM transformer in Figure 5. After finishing the steady operation of solenoid valve the developed valve module to be composed in solenoid valve 2 set and manipulator 1 set is tested. At this time, the starting signal of valve is PWM signal to be formed from microcomputer and solenoid driver. The microcomputer received input of analog signal of joystick and transmitted output of PWM signal to valve module. This PWM signal is changed in

proportional to analog input value of joystick. After testing the valve module is installed on hydraulic actuator.

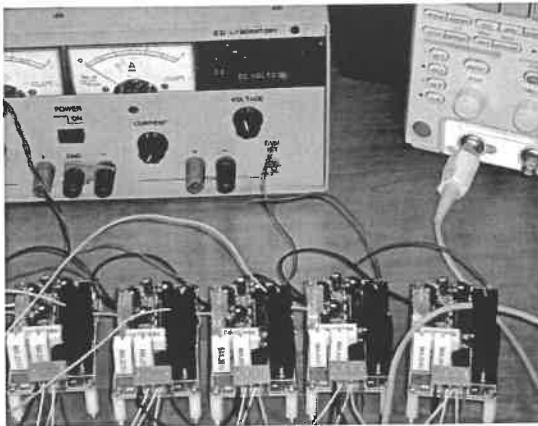


Figure 4. Test of solenoid valve driver

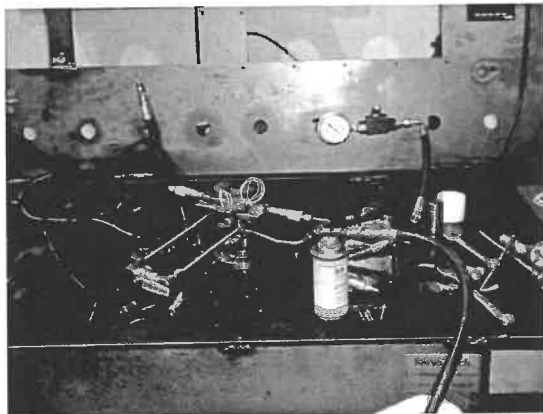


Figure 5. Calibration system of zero position of solenoid valve

Figure 6 shows that the tested hydraulic actuator and valve module are installed on joint link. The working assistant system is operated by muscular amount of human body motion. The first function of the leg part in working assistant system is to realize fundamental characteristics of gait motion. To perform the smooth gait motion as like human body the leg part requires a variety of degree of freedom of hip, knee, ankle, shoulder, elbow, and wrist joint motion. The structure of working assistant system to be assembled with links and joints shows in Figure 6. The size of working assistant system is suitable to be wearable for worker. The hydraulic actuators and control valves are installed on the proto type of structure.

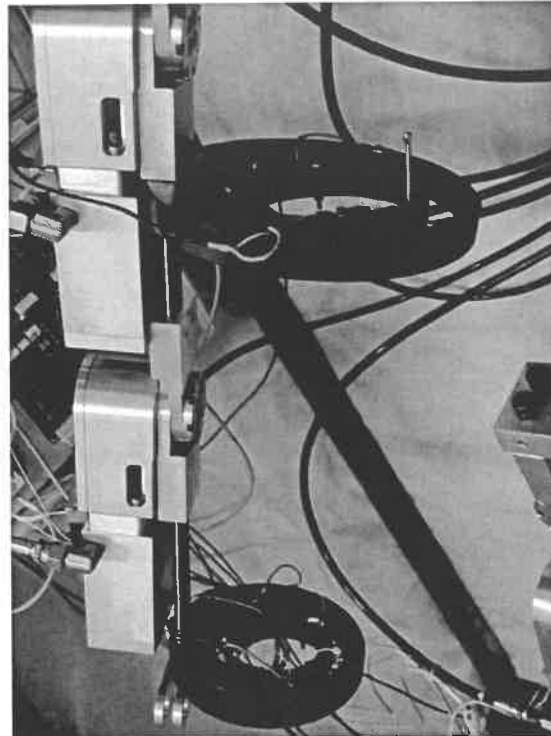


Figure 6. Hydraulic actuator installed on working assistant system

COCLUSION

To improve the productivity of ship building industry the repeated and accumulated fatigue of worker should be reduced in manufacturing process of small and large size of block to be assembled in dock site. In this study the multi-axis link mechanism to smoothly do working motion in a variety of working path is designed and manufactured. The structures of hydraulic actuator and joint link are designed and manufactured to be wearable for worker. In near future the mechanism and control technology of working assistant system to reduce the effect of weight of working tool and working load of working process such as the reaction force of grinding tool will be executed.

ACKNOWLEDGEMENT

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MODELING AND DESIGN OF A MICROSCALE MULTIPLEXED TEMPERATURE CONTROL SYSTEM

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INTRODUCTION

Controlling the temperature of microfluidic sample volumes is essential for a wide domain of applications, including genetic sample preparation using the polymerase chain reaction (PCR) [1-3], cell lysis [4,5], protein denaturation [6], heat shock DNA transformation [7], enzyme reaction control [8], and numerous other chemical and biological procedures with temperature-dependent behavior. Inaccurate temperature control during thermocycling for PCR can diminish efficiency, yield non-specific products, and, in some cases, completely inhibit the reaction. Conventional methods based on thermoelectric devices are typically slow to transition between temperature setpoints, imprecise (e.g. $\pm 1^\circ\text{C}$), and limited to a single temperature for all samples at a given time. Micro-fabricated integrated resistive heaters and temperature sensors [1,9] can achieve improved precision and ramping rates but at the cost of device complexity and surface fouling. A direct, non-contact approach to heating liquid samples with radiation has been demonstrated with simple fabrication requirements and fast ramping rates but performs at low throughput and limited control due to the nature of the broadband, incoherent source [2]. Others have implemented coherent radiation for temperature control of droplets via lasers and fluorescence-based temperature monitoring [10,11] but inconsistent sample size and position limit the system to serial processing of multiple samples.

The aim of this work is to develop a methodology and instrument for controlling the temperature of an array of microfluidic reaction chambers with independent, simultaneous control of individual samples. Optical and heat transfer modeling, precision positioning and fabrication, and an open loop control scheme are applied to produce compact instrumentation that will set new benchmarks for the performance and throughput of microfluidic genetic sample preparation. Here we present a

proof-of-concept for this multiplexed temperature control system.

MODELING

To exploit the advantages of non-contact radiative heating using a laser source and any combination of sample medium and device substrate materials, we have developed a generalized modeling approach [12]. The method begins with determining the absolute radiation absorbed by the sample and surrounding substrate using spectral irradiance data for the laser source, device geometry, and material properties of the target sample. This calculation is then used as a heat generation input for a finite element model of the transient and steady state thermal response, as shown in Fig. 1a.

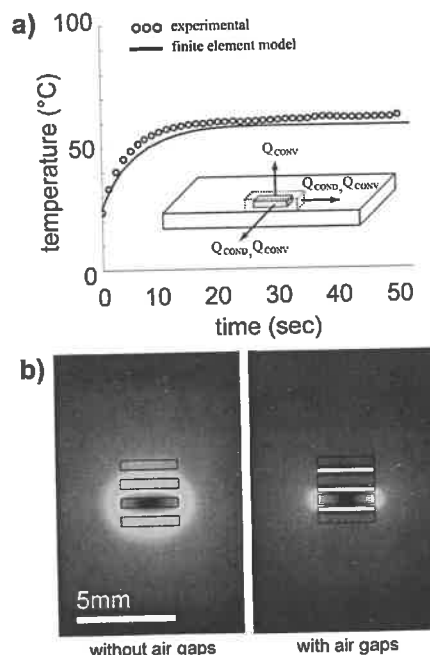


FIGURE 1. Optical and thermal modeling results for heating aqueous samples in microscale chambers. a.) Transient temperature profiles and b.) thermal crosstalk performance can be quickly and accurately assessed.

In addition to using the model for anticipating ramping rates and steady state temperatures, the simulation of thermal crosstalk is essential for confirming the viability of independent control of closely spaced samples. This motivated the addition of air gaps for better thermal isolation, shown in Fig. 1b.

INSTRUMENT DESIGN

Insights gained from the modeling informed the design of prototypes to validate the predicted heating performance and demonstrate a method of multiplexed temperature control. Since scalability is important for the utility of biological instrumentation, an open loop control scheme was devised to obviate the expense and tedium of embedded sensors.

To demonstrate open loop temperature control, an initial single-chamber prototype was built with an optical system consisting of a 1450nm laser diode, selected for the corresponding peak in absorption by water, coupled to a x-y adjustable collimating lens and aligned using a 30mm cage system. This translates vertically along cage rods and is aligned with a polymer microchip holder. A polymethyl methacrylate (PMMA) microchip was fabricated by micromilling and thermal bonding and is used to contain 1 μ L aqueous samples. An exploded diagram and photo of the prototype assembly can be seen in Figure 2.

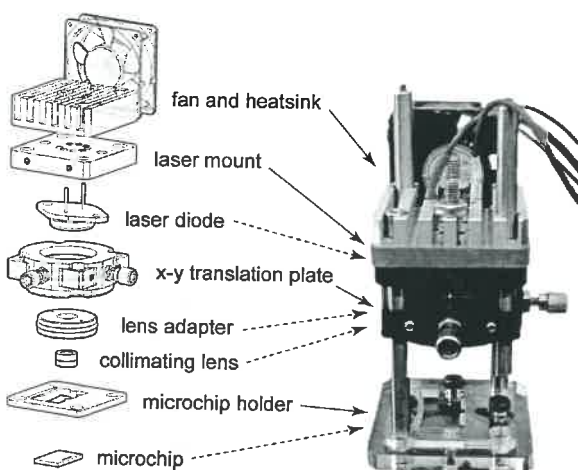


FIGURE 2. Diagram of the first prototype temperature control instrument (left) with the aligned set of optical and microfluidic components. The actual system (right) can be seen in its open configuration for removing or inserting the microchip.

The system was designed for precise and highly repeatable microchip positioning and laser heating, since open loop operation demands consistency to achieve reasonable accuracy. Microchip substrates are first lasercut to exact dimensions and fabrication is performed using a custom fixture aligned to a CNC machine with tools zeroed using electrical conductivity. The microchip holder features an integrated leaf spring and three contact points for full kinematic constraint. Tests performed with a microscope over five trials revealed a standard deviation of microchip position of 5.3 μ m, which results in no measurable difference in thermal response. The laser is equipped with a custom heat sink and fan for stable optical power output.

For calibration, a miniature thermocouple was inserted into the microchip to monitor sample temperature, which was confirmed with a thermal camera. The thermal response for a range of laser power inputs was then recorded. Using the laser driver's transfer function for converting control voltages to power output, a control sequence is written for the desired temperature setpoints of the experiment and parsed into a LabView program that executes the experiment.

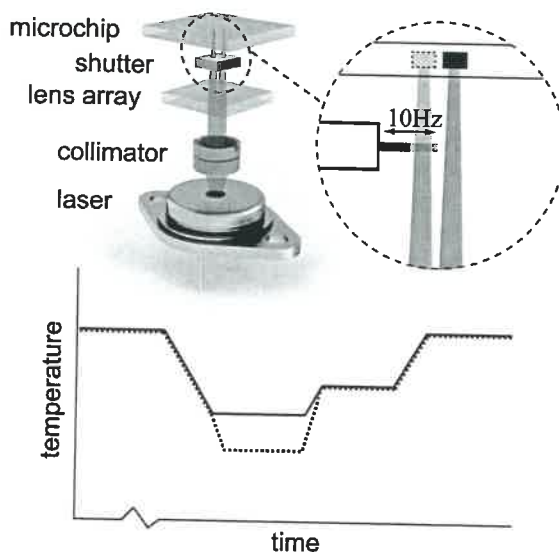


FIGURE 3. Diagram of the second prototype instrument (top) for dual chamber temperature multiplexing. A mechanical shutter operated at 10Hz is used to attenuate the radiation incident on one of the chambers (inset) to achieve independent temperatures, as shown in the representative temperature profile illustrating multiple steps in a thermocycling routine.

To demonstrate multiplexed temperature control, a second prototype was built to integrate a solenoid-driven mechanical shutter (Fig. 3). For this dual-chamber system, in addition to the software-driven modulation of the laser source, secondary modulation via the shutter attenuates the radiation incident on a sample that requires a temperature lower than the setpoint. For reliable optical modulation, the attenuation at a series of duty cycles was characterized for each point in the calibration for a comprehensive mapping of desired temperatures and corresponding control voltages and duty cycles.

Preliminary results (Fig. 4) show that temperature differences beyond 15°C can be maintained between adjacent microscale sample chambers, with limited thermal crosstalk as predicted by the modeling.

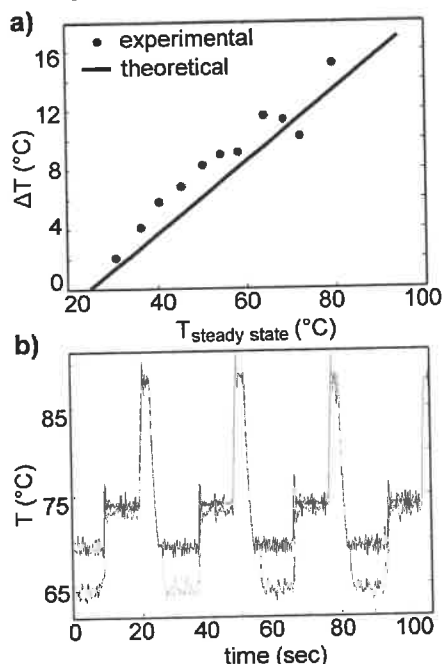


FIGURE 4. Experimental characterization of a dual-chamber temperature control system demonstrates the use of a microshutter to modulate laser radiation and achieve thermal multiplexing. (a) Temperature difference, or thermal crosstalk, ΔT , achievable between adjacent chambers vs. steady state temperature in one chamber, T_{ss} . (b) Thermally-independent cycling measured in our prototype.

CONCLUSION

Combining optical and heat transfer modeling to simulate thermal response of any device with known material properties and a layout amenable to laser irradiation can guide the

design and component selection for myriad applications that rely on temperature control, both existing and enabled by this technology. The simple fabrication requirements for the target device minimizes cost while the coherent radiation with optical modulation maximizes the potential for thermal multiplexing to achieve higher throughputs and diversity of simultaneous conditions.

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POSITION SENSING MODULE BY COMBINING PHOTOGRAMMETRY AND PATTERN PROJECTION

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INTRODUCTION

Photogrammetry extracts data from two-dimensional (2-D) images and maps them into a three-dimensional (3-D) space, reconstructing the object being imaged [1]. The photogrammetric processing is based on the principle of triangulation, whereby intersecting lines in space are used to compute the location of a point in three dimensions. In this paper a low-cost, non-contact technique is described that allows real-time measurement of the location and orientation of a small lightweight module with six degrees of freedom (6DOF). The technique is based on a novel combination of photogrammetry and optical pattern projection. The module generates an optical pattern that is observable on the walls (or other surfaces) of the environment, and photogrammetry is used to measure the absolute coordinates of the features of this optical pattern on the walls. By combining these absolute coordinates with knowledge of the angular information of the optical pattern, a minimization algorithm calculates the location and orientation of the module itself. Thus, the location and orientation of the module is determined by performing photogrammetry on the optical pattern generated by the module instead of carrying out photogrammetry on the module itself. Measurement differences with respect to a calibrated reference of 4 to 5 parts in 10^3 were obtained for a 1 meter range translation test and a 60 degree rotation test.

Figure 1 shows the basic elements of the measurement system, as it might be used to track the position and orientation of the end of a robot, as an example. The module, which consists of a laser diode and an optical projection head, is mounted onto the end of the robot arm. By using the described technique, the motion of the robot arm can be monitored in real time. This indirect measurement offers several advantages. First, the projected optical pattern

can provide a very large number of easily identifiable features for the photogrammetry measurement, which effectively provides an improvement over direct photogrammetry of the module itself through averaging. Second, by using the measurement of the projected optical pattern the measurement process is moved out away from the part, allowing position measurements during manufacturing processes where traditional sensor measurements are difficult or not possible. Additionally, the technique is very inexpensive and potentially has a large measurement range with low measurement uncertainty.

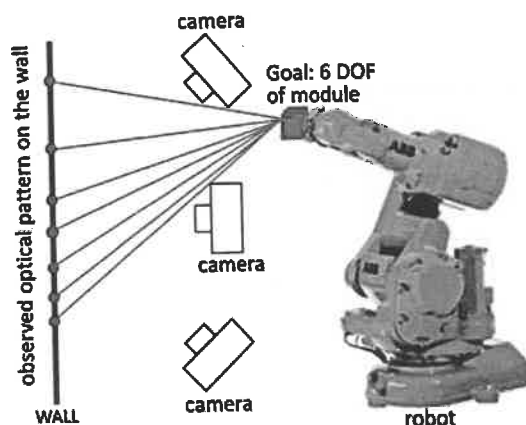


FIGURE 1. The basic elements of the measurement philosophy.

SETUP AND PROCEDURE

The preliminary experimental setup is shown in Figure 2. The module consists of a red laser diode (665 nm, 40 mW) and a projection head. The laser beam propagates through the projection head, and the diffractive element in the head redistributes the laser irradiance into a 7×7 matrix of beams with a full fan angle of $11.4^\circ \times 11.4^\circ$ ($1.9^\circ \times 1.9^\circ$ between adjacent beams). The 7×7 beam array forms a nominally square dot pattern when oriented and observed perpendicular to a wall. The module is mounted

on a 360° rotation stage which is mounted on a precision translation rail that has a resolution of 1 μ m and a translation range of 1.05m. Three 9 megapixel digital cameras (Canon PowerShot SX110IS) are positioned at various orientations with respect to the wall. The cameras are connected to the computer through a USB hub where a remote control software package (PSRemote Multi-Camera® [2]) is used to trigger the cameras to take pictures within 50ms of each other. The images taken by the cameras are processed by a photogrammetry package called PhotoModeler® [3]. A reference length is attached to the wall to provide a known length and a global coordinate system (see inset in Figure 2). The designated Y axis of the reference has a length of 254 $\pm$ 0.1mm. Camera calibration was performed for each camera prior to their use in the experiment. This determines the geometric properties and errors of each camera (focal length, format size, principal point, projection center, etc.), as well as lens distortion.

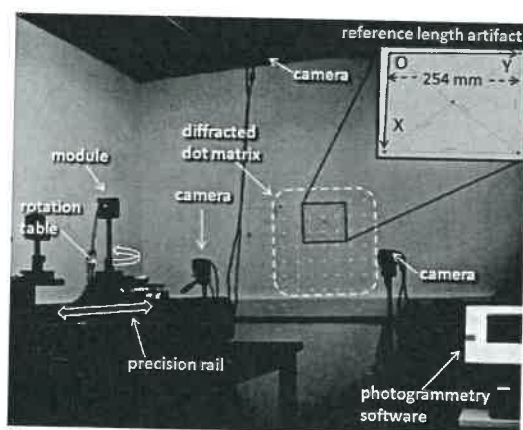


FIGURE 2. Preliminary experimental setup. Inset: defined reference length artifact.

The procedure consists of the following steps : (1) The module is placed at a start point and generates an optical 7 $\times$ 7 dot matrix pattern on the surrounding walls (as shown in Figure 2). The first measurement position is arbitrarily defined as zero. (2) Several images of the observed dot pattern are taken simultaneously with the three cameras and are subsequently processed by photogrammetry to obtain the absolute coordinates of each observed dot. PhotoModeler® [3] is used for the photogrammetry bundle adjustment and the determination of the coordinates. (3) The xyz coordinates, along with details of the projection head geometry are input into a custom chi-square minimization algorithm in MATLAB to

determine the location and orientation of the module. (4) The module is moved in 100mm increments and the steps are repeated.

PHOTOGRAMMETRY MEASUREMENTS AND RAIL CALIBRATION

Because photogrammetry is a crucial part of this technique, the repeatability of individual photogrammetry measurements under our lab conditions was investigated. In this part, steps (1) and (2) of the procedure were repeated ten times to measure dots coordinates on the wall. For the coordinates of the central dot in the 7 $\times$ 7 dot matrix, the standard deviations in x, y, and z are 0.06mm, 0.05mm and 0.04 mm, respectively.

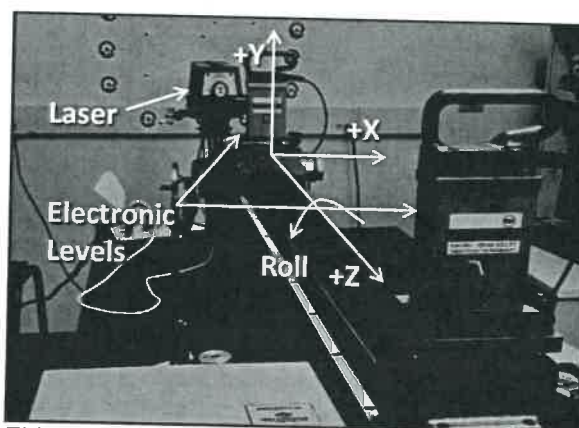


FIGURE 3. Rail calibration setup used in the experiment.

The precision rail was calibrated to determine the positional accuracy by using a laser interferometer following the guidelines in reference [4]. The pitch, yaw, roll, linear displacement, and straightness were measured using a laser interferometer and electronic levels. The coordinate system, as well as the setup, are shown in Figure 3. Based on six repeated measurements over the length of the rail, the mean values of the rail straightness dyz with the standard deviation at each position are shown in Figure 4. The error bars, representing one standard deviation, are less than 1 μ m at most positions. The positive Y direction is defined to point upwards, so results shown in Figure 4 indicate that the rail is lower in the center of its travel.

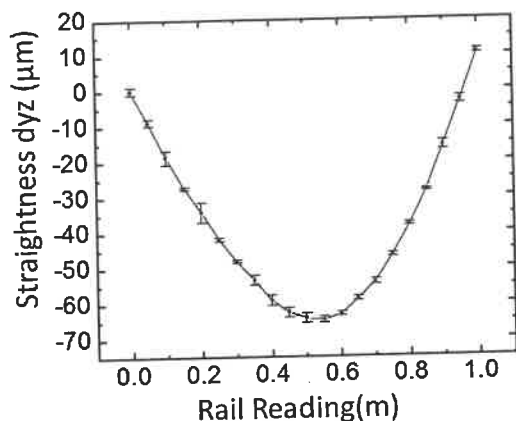


FIGURE 4. Straightness (dyz) of the rail.

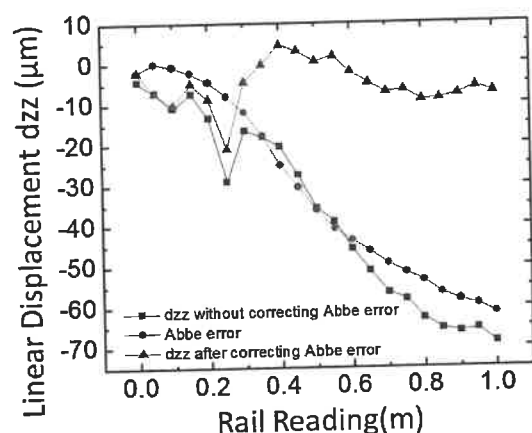


FIGURE 5. Linear displacement (dzz) of the rail.

The raw linear displacement (dzz) data from the interferometer measurements of the rail is the black curve (■) shown in Figure 5. Since the beam path (laser and projection head) in the following experiments was approximately 130mm above the location of the rail encoder, there is an error introduced by this Abbe offset and the rail straightness. This Abbe error is shown as the red curve (●) in the figure. The corrected linear displacement of the rail is shown as the blue curve (▲). It can be seen that the rail encoder reading of the linear displacement of the rail, as measured with the displacement measuring interferometer, is less than 10 μm at most positions. This is

significantly less than the variability observed in our measurements, as seen in Figure 6 (a).

RESULTS AND DISCUSSIONS

With the setup shown in Figure 2, a one dimensional translation test and a rotation test were performed separately. For the translation test, the module was translated 1 meter along the precision rail at 0.1 m intervals and at each position the module location was measured using our method described above. The translated distances are calculated using the coordinates of the module between the current position and the start position, and these position coordinates are in the global coordinate system defined by the axes and scale on the wall. The translation test was done three times and the difference between the measured distance and the rail reading at each position is shown in Figure 6(a). The results show that the measurement differences at all positions are less than 4mm over the 1m range. The measurement errors observed in this initial test of the system are on the order of 4 parts in 10^3 .

For the rotation test, the module was rotated through 60° and measurements were taken every 8° . The measured rotation angle is calculated based on the angle change of the vector from the module to the center dot in the projected dot matrix. This measured angle is compared to the angle reading on the rotary stage. The angle difference, i.e. the difference between the measured angles and the reading from the stage scale, is shown in Figure 6(b). The angle differences are no more than $\sim 0.3^\circ$ for all measurements over the 60° range. The error bars in Figure 6(b) represent the measurement uncertainty, where this is a combination of the measurement standard deviation, u_{measured} , and the reading uncertainty, u_{reading} . The latter uncertainty is our ability to correctly read the scale on the rotation table, this is estimated at $\pm 0.2^\circ$. The uncertainties are combined as described in equation 1:

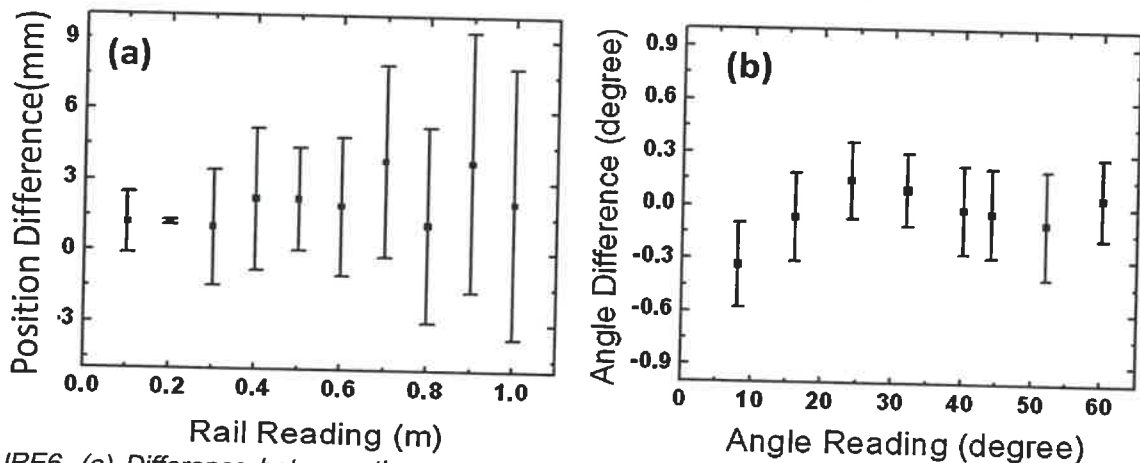


FIGURE 6. (a) Difference between the measured and rail position in the translation test. The error bars represent ± 1 standard deviation over three tests. (b) Difference between angles measured and angles read from the mechanical rotation stage.

$$u = \sqrt{u_{\text{measured}}^2 + u_{\text{reading}}^2} \quad (1)$$

The results show agreement between our measurement and an independent measurement at a level of 4 parts in 10^3 for one dimensional translation and 5 parts in 10^3 for a 60° rotation test. Though only two degrees of freedom were tested in the experiments, the results demonstrate that the technique can be used to trace the motion of the target in all 6DOF. This is possible because the coordinates as well as the vectors from the module to each dot can be solved for at each module position, thus establishing the complete angular orientation of the module in the global coordinate system.

There are a few factors that contribute to our (admittedly large) measurement uncertainty estimates, and are under investigation. These include error in the reference length, error in the angles between the projected beams, the narrow view angle of the projected pattern, and error in the dot coordinates determined by photogrammetry. Work is underway to evaluate each of these factors, and where possible reduce the uncertainty contributed by each factor. After the evaluation is done, feasibility tests of the system will be performed using a robot arm.

CONCLUSIONS

In this paper we describe a technique that combines photogrammetry and optical pattern

projection to determine the location of a sensing module. Experimental results show agreement at a level of 4 parts in 10^3 for the translation test, and 5 parts in 10^3 for the rotation test. Numerical simulation results [5] show that our proposed technique can achieve lower measurement uncertainty than photogrammetry alone by increasing the number of diffracted beams in the projection pattern and increasing the full view angle of the projected pattern.

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RETROFITTING A HAAS GT-20 LATHE WITH AN OIL HYDROSTATIC SPINDLE

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BACKGROUND

This paper covers our experiences modifying a Haas CNC lathe with a model 4R-H, step compensated, hydrostatic spindle. Professional Instruments Company is well known as a manufacturer of precision spindles, but we also do high precision subcontract machining for a variety of customers. One of our biggest customers requires large numbers of precision lens holders that have sub-micron tolerances on roundness, flatness and concentricity. In the past, this level of precision could only be achieved by grinding on air bearing equipped machines, but with the number of parts required that was determined to be impractical.



FIGURE 1. The Haas GT-20 was chosen because it is inexpensive and is easily modified.



FIGURE 2. The 4RH was designed using the same mounting bolt circles as its air bearing cousin. The compact design makes it feasible to mount in front of the existing Haas spindle, simplifying the spindle retrofit.

SPINDLE RETROFIT

A Haas GT-20 CNC lathe was chosen, because as a "toolroom" type lathe it doesn't have a turret, which simplifies its operation and makes the project much more affordable. We drive the hydrostatic spindle with the original Haas spindle, using a flexible diaphragm coupling. This gives us the ability to freewheel our spindle, while dialing in parts, simply by disengaging the collet. The spindle is mounted on a fixture attached directly to the Haas seal ring, which was modified with 6 tapped holes and reground to allow it to be used for mounting our hardware.

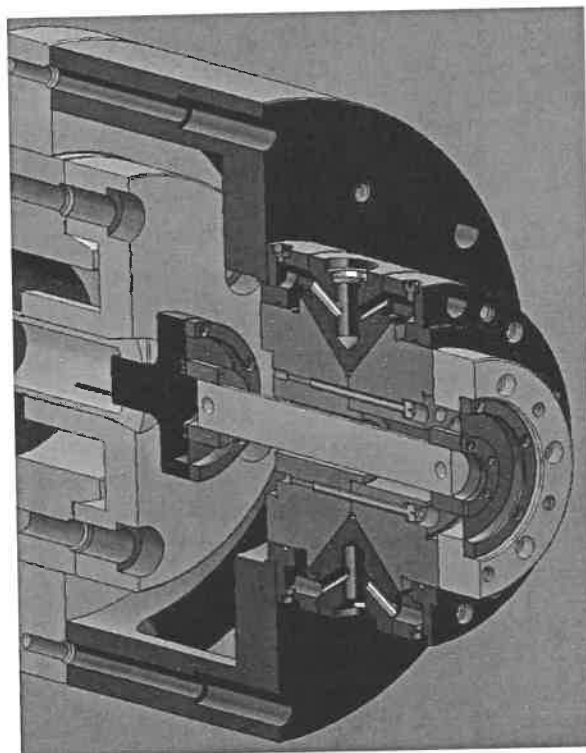


FIGURE 3. SolidWorks rendering shows details of the installation of the 4R-H spindle and diaphragm coupling drive.

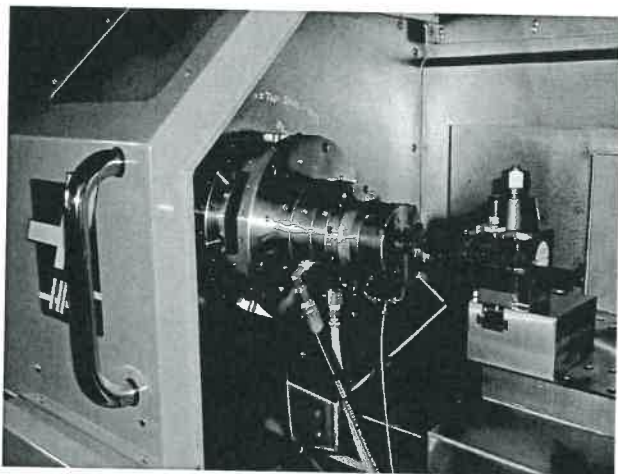


FIGURE 4. Close up of the spindle retrofit. Note: For this test we shimmed the spindle to correct for alignment. Later we ground the mounting hardware to achieve better squareness.

CUTTING TESTS

To test the real world accuracy of our machine, we turned a 150 mm diameter, hardened, (Rc-40), 416 stainless steel test piece, using a Tungaloy CNMG431-TSF AH120 insert. The tool has an angle of 80 degrees and a nose radius of .016". The spindle speed was 600 RPM with a 0.0005 inch depth of cut and a feed rate of 0.000625 inches per revolution.

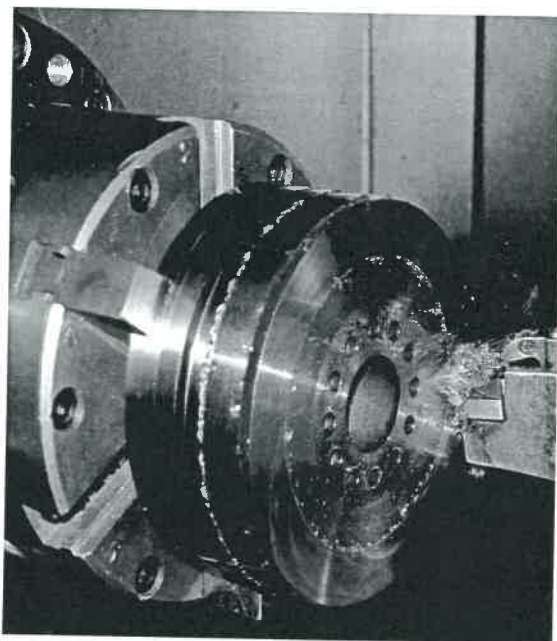


FIGURE 5. Actual part being cut on the Haas GT-20.

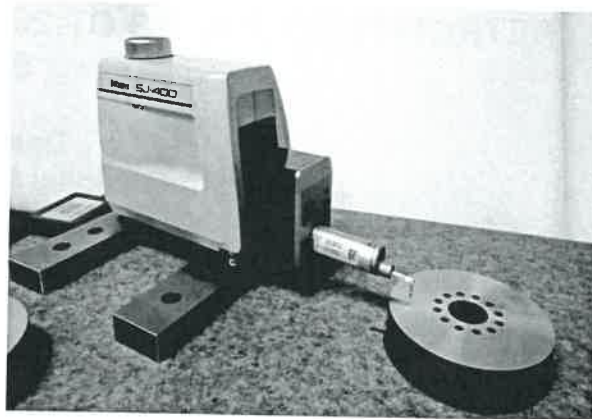


FIGURE 6. Checking surface finish on a finished part.

Ra 0.11 μ m
 Rz 0.8 μ m
 Rq 0.14 μ m
 R-PROFILE
 EVA-L 4.0mm
 $\lambda c=0.8mm \times 5$
 $\times 10K$
 $\times 50$
 Ver. 1.0 μ m/cm
 Hor. 200.0 μ m/cm

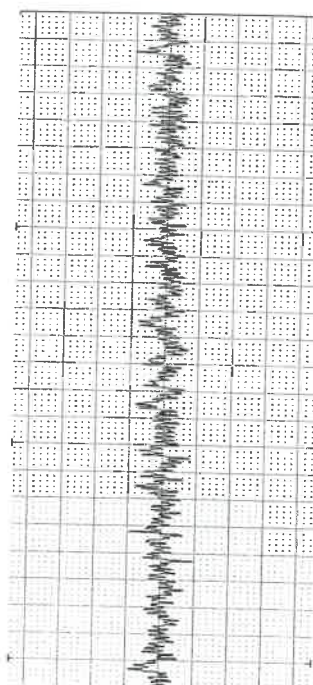


FIGURE 7. Surface finish results of a facing test cut. 4 micro-inch Ra.



FIGURE 8. Air bearing roundness tester for checking roundness and circular flatness, uses software developed by Professor Eric Marsh at Penn State.

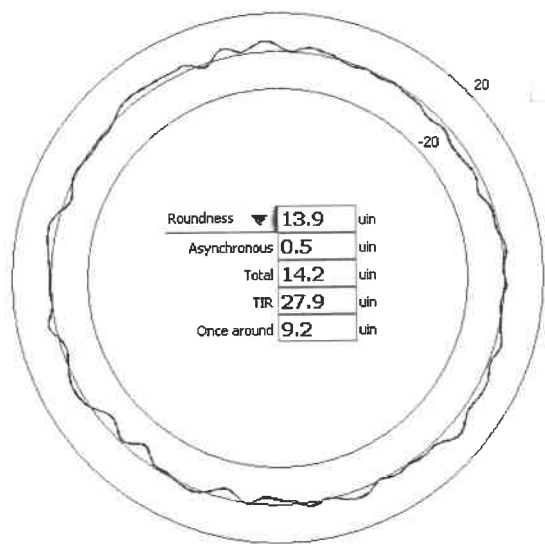


FIGURE 9. Roundness chart of a hardened 416 stainless steel test part, turned on the 4R-H, in the Haas GT-20.

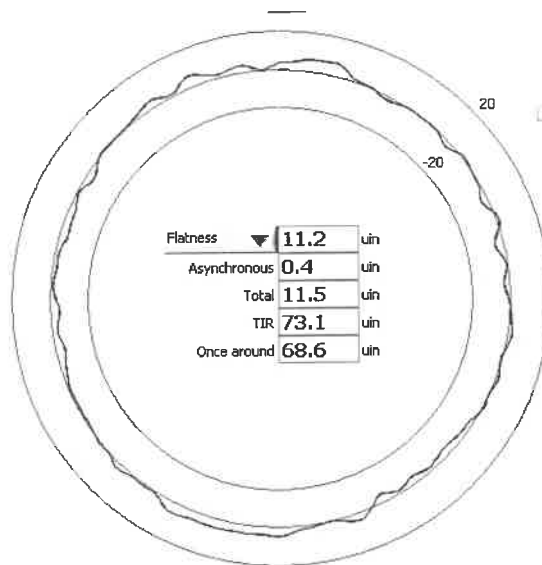


FIGURE 10. Circular flatness of part measured on air bearing roundness tester.

CONCLUSIONS

In hard turning applications, the superior load capacity and dynamics make a hydrostatic spindle a better choice than air bearings. Furthermore we have been pleasantly surprised at the level of accuracy in an inexpensive Haas lathe. We are planning more extensive tests in the future, where we will be cutting hardened tool steel.

EFFECT OF JOINT STIFFNESS ON THE DYNAMIC PERFORMANCE OF MACHINE TOOL

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INTRODUCTION

Machine tool performance has many influencing factors from feed rates, spindle rotation, various systems that turn on and off during the machining operation, thermal coefficients, and the stiffness of the machine tool and more. Joint stiffness is one factor that is typically overlooked in evaluating the performance of machine tools. This paper presents the effect of the joint stiffness on the dynamic performance of the machine tool by using the frequency response function (FRF) of the tool-spindle system, structure, and measurement of vibration (power spectrum) during heavy milling, and analysis using the stability lobe diagrams (SLD). These methods of measurement show how non-uniformed scraped surfaces and improper threaded holes are the cause for weak joint stiffness and poor machine tool performance.

MACHINE TOOL CHATTER PROBLEM

Machine tool manufacturers typically will perform an in-house test that increases the amount of material removal beyond normal specifications to investigate the shape and surface accuracy on the workpiece to determine if the machine tool is working properly prior to shipping. The bridge type machine tool in our case study is designed with a high material removal rate (MRR) but exhibits chatter in the workpiece. The milling chatter occurs when the spindle head is at the position (Z = -800mm) and only in one feed direction (-X). The specifications of the machine tool and the milling conditions are listed below.

Machine tool main specifications:

X / Y / Z travel: 2200 / 2300 / 920 mm

Spindle: Gear type / BT50 (6,000 RPM) / 22KW

Milling conditions:

Spindle speed: S = 500 RPM = 8.33Hz,

Feed rate: F = 1500mm/min, Ft = 0.375 mm/flute

Tool: Sandvik R200-140Q40-20M ϕ 160mm, RCKT2006M0-PH4240 with 8 flutes

Workpiece material: S45C,

Axial depth of cut (DOC): 5 mm,

Radial width of cut (WOC): 100mm

Z axis position: -800mm

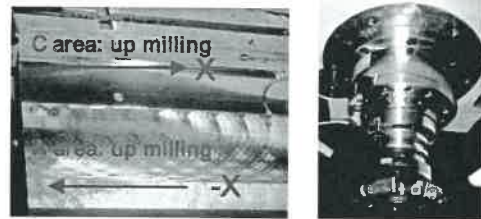


FIGURE 1. Tool marks generated on the workpiece are from the tool itself and two different feed directions.



FIGURE 2. Bridge type machine tool. (Courtesy of Vision Wide Tech Co., Ltd.)

FRF TEST ON SPINDLE AND STRUCTURE

Equipment used: 1-axis accelerometer (Endevco 27AM1-10), 3-axis accelerometer (Kistler 8762A50), small modal impact hammer (Kistler 9726A5000), big modal impact hammer (Kistler 9728A20000) and 4 channel real time FFT analyzer (Benstone impaq Elite). The FRF of the tool-spindle system is measured by the 1-axis accelerometer which is placed directly on the tool itself and the small modal hammer strikes the tool tip on the opposite side. The measured results of the FRF as displayed in the graph (fig. 3) shows that the tool-spindle system has high natural frequency at 609Hz (X), 591Hz (Y), and 308Hz (Z).

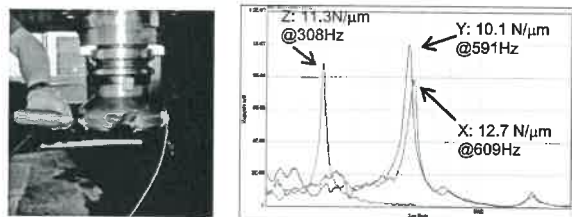


FIGURE 3. FRF of the tool-spindle system.

The FRF of the structure is then measured by using the 3-axis accelerometer, a big impact hammer and FFT analyzer. The 3-axis accelerometer is placed directly on the spindle housing and the modal hammer strikes the spindle housing on the opposite side. The measured results of the FRF as displayed in the graph (fig. 4) shows that the structural stiffness of X-direction is lower than the Y-direction or Z-direction.

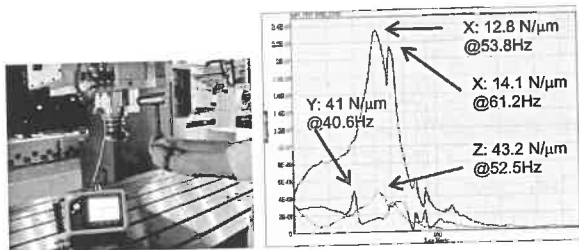


FIGURE 4. FRF of the structure at Z=-800mm.

The spindle position is then moved from Z= -800 mm to Z= -700 mm, and Z= -650 mm, and the test repeated. The X-axis first natural frequency of FRF changes from 53.8Hz to 54.1Hz and 56.3Hz respectively. The dynamic stiffness ($2k\zeta(1-\zeta^2)^{1/2}$) and the damping ratio changes dramatically, indicating that the dynamic stiffness of the structure is sensitive to the Z axis position (fig. 5). The X-axis dynamic stiffness of the first natural frequency is 12.8 N/μm at Z= -800mm (fig. 4).

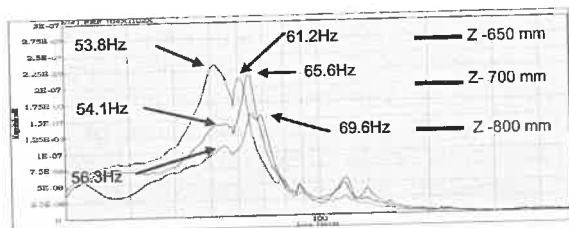


FIGURE 5. X-axis FRF of machine tool structure at Z= -650mm, Z= -700mm, Z= -800mm.

STABILITY LOBE DIAGRAM (SLD)

The stability expression was first obtained by Tiusty [1]. Tobias [2] presented a similar solution. Altintas [3] did much more detailed research in the field. Stability lobe diagrams show the maximum depth of cut that you can expect from the machine tool. The SLD and cutting force graphs are obtained by using CUTPRO software [6] (Manufacture Automation Laboratory Inc.) by inputting the measured FRF readings. For high speed machining or heavy machining, if the dynamic stiffness of the structure is higher than

the tool-spindle, then the cutting chatter is dominated by the tool-spindle dynamic stiffness, called regenerative chatter. So, we consider firstly the tool-spindle system FRF for evaluation using the SLD.

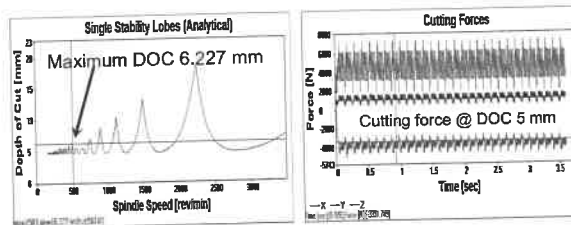


FIGURE 6. SLD of the tool-spindle system FRF only - showing expected DOC of 6.227 mm and milling force of 5 mm DOC milling (no chatter). [6]

A machine tool very often has structural stiffness greater than 50N/μm. However, our results show a weak structural dynamic stiffness. When we add the structural dynamic stiffness together with the tool-spindle FRF to create a new SLD, it shows the expected maximum depth of cut to be reduced to 3.138 mm (fig. 7).

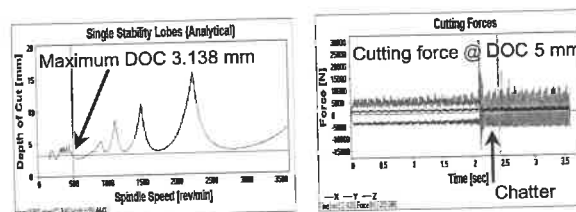


FIGURE 7. SLD of 3.138 mm of the structure and spindle. Simulated chatter observed at 5 mm DOC cutting force. [6]

This tool-spindle system is estimated to be capable of milling a 6.227 mm DOC without chatter, but we cannot achieve this depth of cut. The estimated maximum DOC is only 3.138 mm for the structure (fig. 7). Our conclusion is based on the SLD and FRF analysis that the chatter problem is a result from the structural dynamic weakness, not from the spindle.

MILLING AND VIBRATION TEST

To perform the vibration spectrum measurement, a 3-axis accelerometer is placed on the spindle head with a 5 mm depth of cut and 100 mm width of cut milling, and the results recorded. A tooth passing frequency is observed of (66.7Hz) and a chatter frequency of (51Hz) which is excited when the chatter vibration occurs in the A area (up milling).

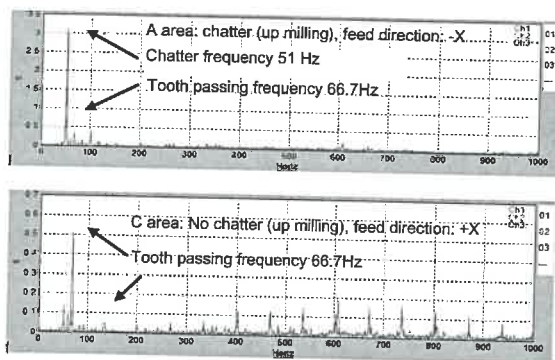


FIGURE 8. Vibration spectrum as milling test with 5 mm DOC & 100 mm WOC.

Notice that there is no chatter in the C area. The vibration power spectrum shows the tooth passing harmonic frequencies (fig. 8). The milling force is the same in these two areas. But the force direction acting on the spindle is opposite (fig. 9), which means the direction of force changes the structure dynamic stiffness. This is another indication of a weak structure.

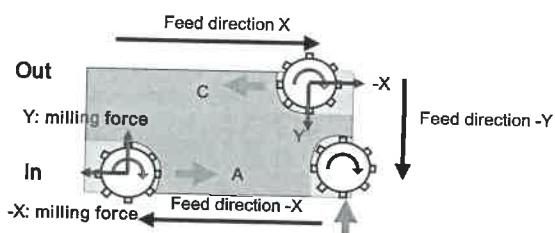


FIGURE 9. Milling force and direction acts on the spindle. (A area: milling-force pulls the spindle out from structure, C area: milling-force pushes the spindle into the structure.)

MODAL TESTING

Modal testing is a proven method to find the location of the mode shapes on the machine tool. The input force and the model are shown in Figure 10. The FRF data is shown in Figure 11.

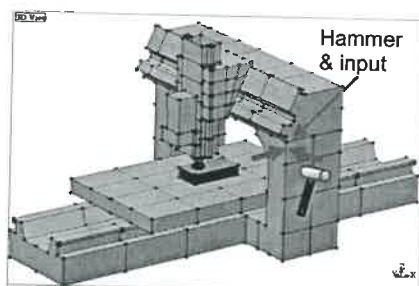


FIGURE 10. The measuring points and force input of modal testing. [7]

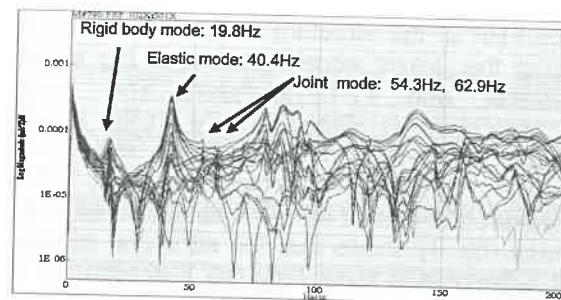


FIGURE 11. The FRF and joint mode frequencies as shown on the plot.

The natural frequency, damping ratio and mode shapes are estimated by using ME'scope software [7] based on the modal values obtained from the Benstone Instruments' Impaq elite analyzer. They are listed in Figure 12.

Mode	Frequency (Hz)	Damping ratio (%)
1	19.8	2.1
2	40.4	1.8
3	54.3	13.1
4	62.9	7.7
5	78.9	1.5
6	85.6	1.6
7	92.5	1.4

FIGURE 12. The natural frequencies and damping ratios are shown in the table.

The 1st mode (19.8 Hz) is rigid body motion. The 2nd mode (40.4 Hz) is an elastic mode. The double columns twist in this mode. The 3rd mode (54.3 Hz) and 4th modes which have high damping ratio (13.1%) are joint separation motion (fig. 13). The damping ratio is normally below 5% for the structure of a machine tool. The 3rd mode (54.3 Hz) is the cause of milling chatter in this case study (very close to 53.8 Hz as measured with FRF and vibration power spectrum 51 Hz). Opposite direction of milling force causes the joint to be stiffer or softer. That motion also changes the dynamic performance, the FRF and the SLD.

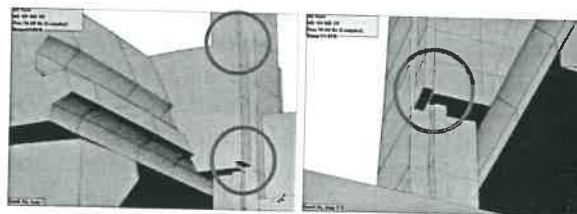


FIGURE 13. Mode 3 (54.3 Hz, damping ratio 13.1%) is the joint separation motion between the spindle head and the saddle. [7]

DISCUSSION

The spindle head is installed into the hard rail of the saddle. The joint interface includes the bolt, thread, and the scraped surface of the hard rail. If the scraped surface is not uniform or is configured with inadequate contact points, plastic deformation will occur in the contact points when force is applied during machining. Therefore, joint stiffness and heavy milling ability are reduced. The weak joint stiffness is sensitive to the amplitude of milling force. Because the contact area is not uniform a crater lip is formed near the thread upon tightening and milling force (fig. 14).

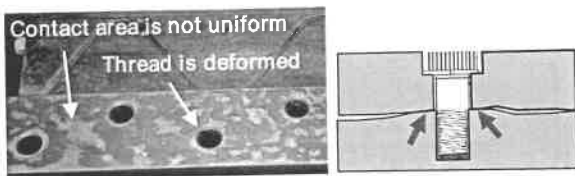


FIGURE 14. The scraping surface is not uniform and the tightening force form a crater lip.

To correct this problem, threads are redesigned to start a distance below the surface to avoid forming a crater lip upon bolt tightening and during milling force. Both surfaces of the saddle are finely re-scraped to achieve a more uniform surface and thus proper contact points. The rule of scraping and bolt joint are presented by Moore [5] and Slocum [6].

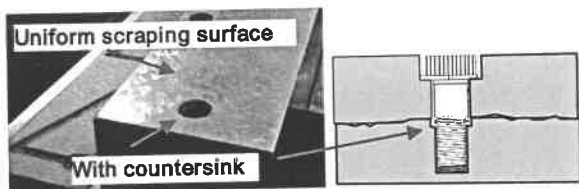


FIGURE 15. Rebuild the scraping surface and the countersink to get a stiffer joint.

The FRF of the rebuilt joint has a much higher dynamic stiffness (fig. 16). The dynamic performance is also improved in SLD (fig. 17).



Figure 16. The new FRF is stiffer than the previous one.

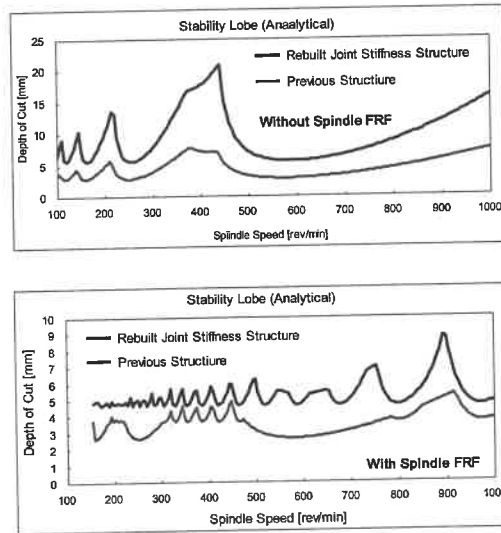


FIGURE 17. The SLD of the rebuilt joint structure shows a higher depth of cut. [6]

CONCLUSION

This paper describes the approach of using frequency response function (FRF), power spectrum measurements and stability lobe diagrams (SLD) to clearly identify weak joint stiffness that contribute to machine tool performance during machining. Using these techniques also help understand the amount of improvement achieved after correcting the discovered problems.

ACKNOWLEDGEMENT

In accomplishing this work, we acknowledge the helpful support of Professor Altintas and the Vision Wide Tech Co., Ltd.

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HYBRID SYSTEMS FOR HIGH ACCURATE TRAJECTORY TRACKING WITH SUB NANOMETER RESOLUTION

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INTRODUCTION

Nanotechnology along with semiconductor production and inspection tools increasingly require motion control with longer travel ranges combined with nanometer or even picometer resolution. Pure piezoelectric stages and standard motorized positioning systems with spindles are not able to meet such demands. For a long travel range actuator, ultra high resolution air bearing or hydrodynamic guiding systems are usually implemented to ensure the required resolution. The capability of such friction less guiding systems are limited when high load capacity compact systems are required.

A mechatronic hybrid approach with high quality closed loop control of the resulting complex system structure can provide an alternative. Excellent trajectory tracking can be achieved even in compact mechanical designs. This article describes our experience in hybrid positioner design for accurate tracking control. Simultaneous control of piezo flexure or voice coil flexure actuators together with motorized drives provide inches of travel with nanometer or even sub-nanometer resolution.

HYBRID MULTI ACTUATOR SYSTEM DESIGN

Stages with piezo actuators achieve high resolution in many applications with high stability and fast settling time. They are however limited in stroke. They typically do not exceed a few hundred microns while remaining within reasonable dimensions. The advantage and high stiffness of piezo actuators also disappears with the square of the lever transfer ratio. One alternative are piezo walking drives, where the piezo expansion effect is stacked by a controlled combination of clamping and expansion. In the early 1990's we developed a walking drive technology based on four separate piezo actuators. [1]

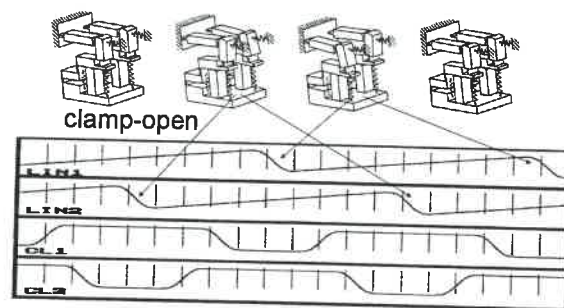


Fig.1: Piezo walking drive with two piezo driven legs and two clamp-open piezo stacks

The newer NEXLINE® piezo walking drive technology, developed at the end of the 90's, is a piezoelectric hybrid system combining the d33 longitudinal piezo effect with d15 shear deformation to achieve both, clamping and expansion. All these walking drives require high resolution sensors and digital controllers for both step control and analog control of the hybrid piezo stacks.

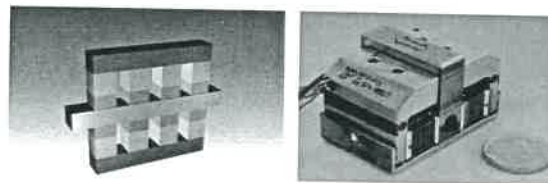


Fig.2: NEXLINE piezo walking drive with longitudinal and shear piezo

With powerful new digital controllers, we can compensate nonlinearity and hysteresis effects in the piezo sub-modules and improve the linear motion quality of such walking drive actuators. Clamping and pushing forces are also increased and a velocity up to millimeters per second can be achieved. Walking drive actuators with pushing forces of more than 400N are now available.

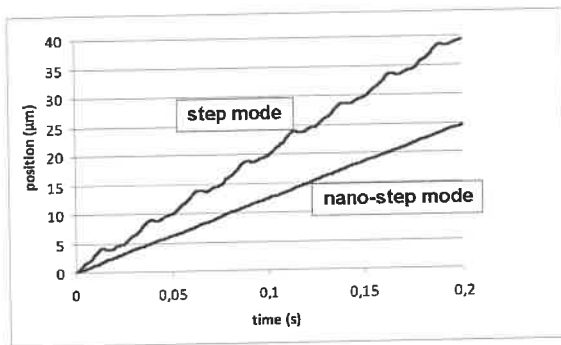


Fig.3: Trajectory tracking smoothness of piezo walk drive systems with hysteresis compensation and usage of high resolution incremental sensor

HYBRID ACTUATOR WITH TWO DIFFERENT ACTUATOR PRINCIPLES

For higher load conditions (hundreds or thousand of Newtons) and higher speed (some inch/sec) a hybrid design of traditional motorized spindle system and piezo actuator has been developed. This aims to overcome the limitations of traditional systems which are mainly caused by spindle friction and surface roughness producing distortions in motion.

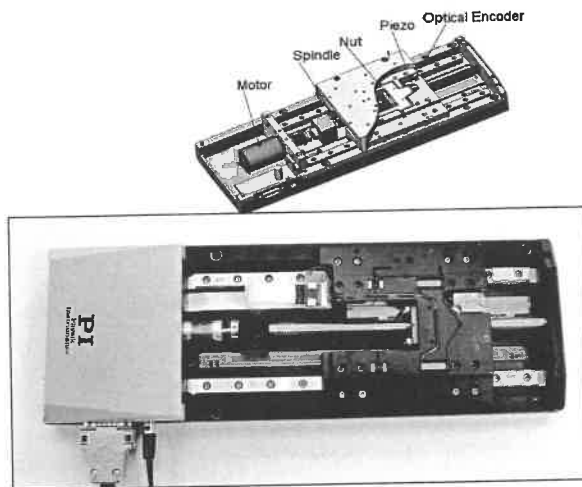


Fig.4: Hybrid stage with electric motor and piezo actuators

CONTROLLER HARDWARE STRUCTURE

One challenge with a hybrid system is designing a controller based on an accurate incremental sensor which can implement a closed loop servo (a cascade servo control loop) using a piezo or voice coil actuator to correct the motion

distortions of the spindle drive. Our controller hardware structure was described in [4]. Recently we have developed a new three axes controller based on Q7 PC- modules, internal PCI-express bus and clock synchronization with FPGA's.

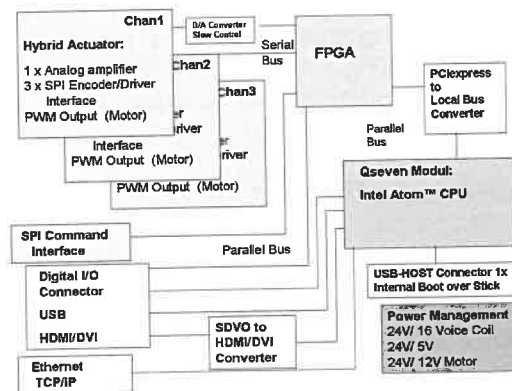


Fig.5: Controller hardware structure

CONTROL STRUCTURES FOR MULTI ACTUATOR HYBRID STAGE

Dominant transfer functions:

The transfer function of a two actuator hybrid stage is dominated by the transfer function of the actuator directly coupled to the motion platform. In our implementation, that is the piezo-flexure or voice coil-flexure part of the structure and it should determine, together with the mass of any load, the systems main resonant frequency.

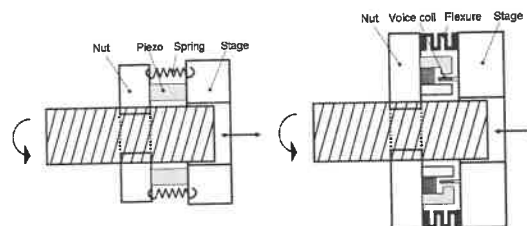


Fig.6: Piezo with preload and voice coil in flexure

If in hybrid systems voice coil actuators are used as fine positioning systems they have to be built into flexure assemblies to prevent low stiffness in open loop operation. In this case, the voice coil assembly (working on a friction less flexure) will determine the main structural resonant frequencies.

The alternative use of piezo actuators provides a predefined high stiffness due to the PZT piezo ceramic material. In some cases it is desirable

for the open loop (passive) stiffness of hybrid systems to have a medium stiffness, enabling additional damping of external vibrations. Voice coil actuators in hybrid systems have the advantage of variable stiffness. In typical designs both types of actuators will have about the same actuation stroke but the voice coil actuator will operate in a larger range of the deformed spring. Voice coil actuators are suitable for millimeters of working range with micrometer resolution.

Control loop with stroke limitation

The high resolution piezo – or voice coil actuator is stroke limited. The main control loop is strong dependent on the fine actuator, e.g. the short distance actuator.

The servo control loop for piezo actuators can be stabilized with a simple PI (proportional integral term) filter. This filter's transfer function can be described with:

$$T_c(p) = K_r \left(1 + \frac{1}{T_i \cdot p} \right) \quad (1)$$

T_i integral time constant

K_r gain of the servo filter

For short stroke voice coil actuators with current loop the servo filter needs a PID term:

$$T_c(p) = K_r \frac{1 + (T_i + T_g) \cdot p + (T_i \cdot T_g + T_i \cdot T_d) \cdot p^2}{T_i p + T_i T_g \cdot p^2} \quad (2)$$

T_d differential term

T_g smoothing term

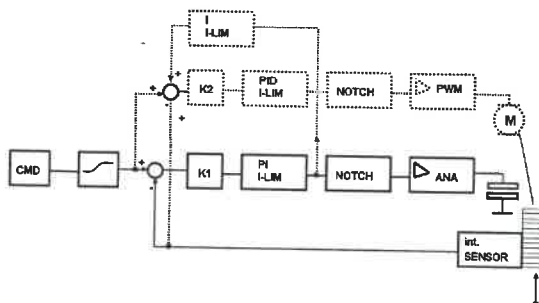


Fig. 7: Main loop via short distance actuator

The control input for the second loop is created by the output of the PI-term of the short distance actuator. An additional I-term (with I-limitation) increases the tracking performance.

To reduce ringing around the resonant frequency an additional notch filter is

implemented at this frequency. This allows the use of a high proportional gain.

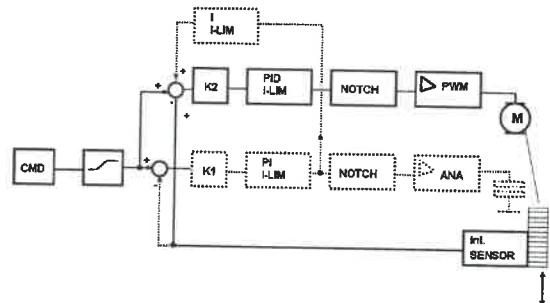


Fig. 8: Course loop of the hybrid controller drives an electric motor

In the last years some different hybrid and control strategies have been developed. Fujita et al. [2] describe a system where the short actuator is located away from the stage with 5nm resolution. Juhasz et al. [3] show that a fast and precise current loop, a piezo hysteresis compensation module and a state space control algorithm can increase the trajectory tracking performance but the algorithm is very complex and needs extreme high computational power.

Controller with external sensor

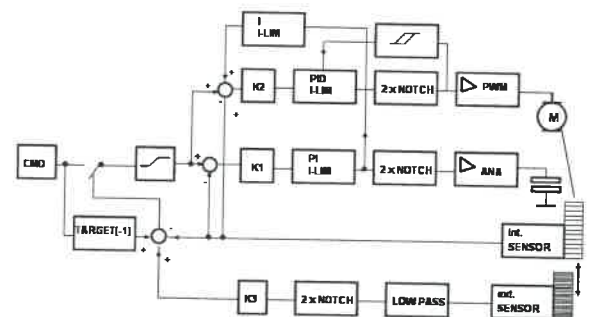


Fig. 9 Hybrid control schematics with additional external sensor (incremental optical or magnetic sensor or interferometer)

With this design, the stage motion is measured directly by the incremental sensor. For long stroke stages there is no alternative to incremental sensor systems as the signal to noise ratio of analog sensors would limit the hybrid system resolution. PI's capacitive displacement sensors have a signal to noise ratio of better than 100dB or about 200 000 : 1.

Figure 9 describes the configuration of the internal and external sensor setup. The external sensor input moves the target position for the trajectory generator so that the main control loop

is not destabilized. This design provides stability even if highly resonant equipment is placed on the stage. Using a laser interferometer placed in the object plane, the stage can be operated parallax-free without Abbé error misalignment. With this arrangement we can realize systems with 0.6nm stability in the object plane.

TEST RESULTS OF HYBRID STAGES

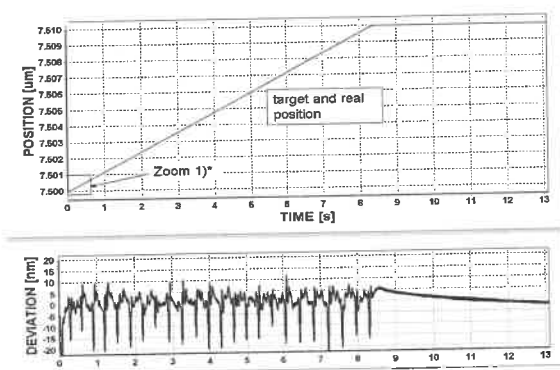


Fig.10: Trajectory tracking of the motor part of the hybrid system (piezo part switched off).

Small stick-slip effects in the spindle system lead to tens of nanometer distortion, when only the motor is used.

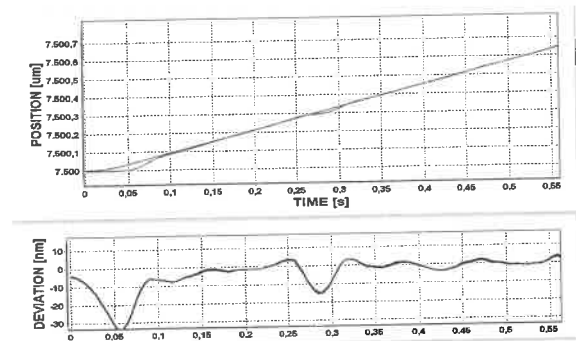


Fig.11: Zoom cut of fig.10, (see 1)* The RMS tracking error is about 5nm

In hybrid systems the servo motor and the piezo are actuated at all time (fig.12). Piezo voltage is actively reduced close to zero during tracking (+/- 36 V maximum).

The hybrid system in fig. 12 shows no significant hysteresis at the stage platform. [4]

Internal and external sensors with resolution significantly better than 1 nanometer can be used in the hybrid system. An actuator we designed for the E-ELT telescope project used a

resolution of 0.125nm. The 24 bit D/A circuitry provide piezo resolution in the picometer range.

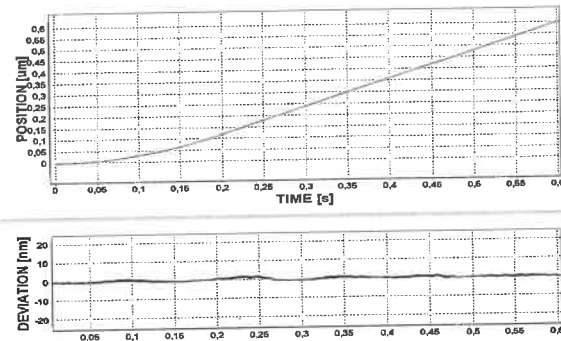


Fig.12: Trajectory tracking of a hybrid system. The tracking error is less than 0.6nm RMS

SUMMARY

Hybrid actuator systems consisting of traditional electric motor drives combined with piezo or voice coil short stroke actuators provide very high resolution motion and positioning. The extreme high resolution results from the well guided short stroke actuator. The simultaneous control of such multi actuator systems requires cascade control loops or mesh coupled state space control. With a single high resolution incremental encoder in the control loop of both actuators the systems can achieve sub-nanometer resolution and stability.

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THE MEASUREMENT OF MESO-SCALE TORSION SPRINGS

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INTRODUCTION

The characterization of meso-scale torsion springs is an important activity for applications relying on the miniaturization of mechanical mechanisms. Designers continue to push the design limits for springs, and are faced with significant uncertainty in the quantification of the static, dynamic and fatigue behavior of small torsion springs. While static stiffness is the primary parameter specified in designs; providing in-situ measurements of dynamic stiffness, fatigue life and operational torque loads would give designers information necessary for further optimization and improved reliability. Because commercial systems do not exist that meet the requirements for range, accuracy or dynamic response, this paper will describe continued work to develop a measurement system for the characterization of meso-scale torsion springs.

MESO-SCALE TORSION SPRINGS

Figure 1 shows an example of a spring targeted by the new measurement system. They are produced through traditional coil winding processes and typically consist of drawn Elgiloy, Hastelloy, Inconel or stainless steel wire. Coil diameters and lengths are typically 4mm or less with wire diameters spanning from 50 to 500 μ m. Mounting methods vary with the design application, so end geometries may have curved or straight tangs with radii of curvature and bend angles in any orientation. Operational torque loads are typically 0.5 to 2.0mN·m, with initial preloads in the range of 0.5 to 1.5mN·m. Normal mechanism motions rely on 20 to 30° of spring rotation beyond their initial preloads at cycle rates up to 100Hz.

MEASUREMENT SYSTEM REQUIREMENTS

While systems for measuring torsion springs exist¹, no commercial equipment meets the requirements for meso-scale torsion springs. Fundamental shortcomings of commercial systems are their inability to measure dynamic spring motions or to quantify torque below 10mN·m. Additional problems with

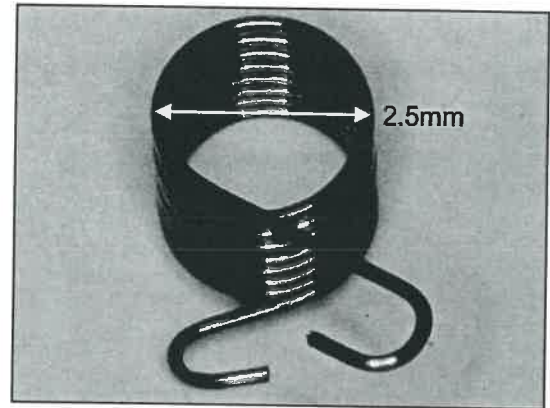


FIGURE 1. A representative meso-scale torsion spring. The wire diameter is 0.175mm, the spring length is 3.8mm, the coil diameter is 2.5mm, and the wire material is Elgiloy.

measurement accuracy, with traceability and with system reliability have been observed with customized equipment delivered to SNL. The presented work details a system developed to address each of these weaknesses. The prototype system has been designed to measure in-situ torque ranging from 0.1 to 10mN·m, at frequencies up to 100Hz and with dynamic rotations of $\pm 15^\circ$. Such requirements enable the measurement of both static and dynamic spring rates. As a result, the effects of resonance, cyclic loading, fatigue, end effector geometry and even defects on operational performance should be quantifiable.

SYSTEM DEVELOPMENT

The prototype measurement system, Figure 2, was designed to be simple, and yet to provide the functionality necessary to meet the design requirements². Recent developments have focused on system improvements thru the elimination of noise sources in the torque sensor signal, the addition of a preload flexure to the voice coil stage and the implementation of closed loop control of the voice coil actuator.

Torque Sensor

Torsion loads on springs under test are measured using a Kistler 9329A torque cell and

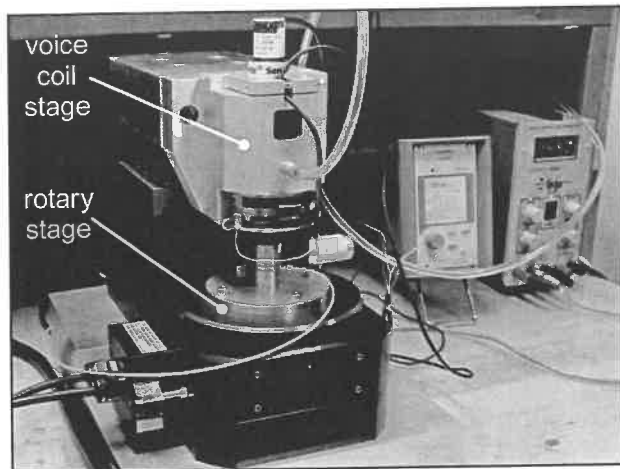


FIGURE 2. The prototype system for the measurement of meso-scale torsion springs.

5015A charge amplifier. Although the 9329A provides a measurement range up to $\pm 1\text{N}\cdot\text{m}$, its lowest range setting ($\pm 10\text{mN}\cdot\text{m}$) is required to measure meso-scale torsion springs with adequate resolution. Cable management was found crucial in this operating regime. Without strain relief, the sensor cable stressed the torque sensor during rotary stage motion and create signal errors of $1\text{mN}\cdot\text{m}$ or greater. Constraining the sensor cable to the table of the rotary stage eliminated stresses on the sensor and reduced the noise threshold during rotation to within the $0.03\text{mN}\cdot\text{m}$ product specification. Initial work also revealed the ability of a magnet to introduce false torque signals, prompting concerns regarding potential errors from operation of the voice coil. Operational tests demonstrated, however, that no measureable error signal was introduced by the voice coil although an offset due to the stator magnets is nulled during spring testing.

Preload Flexure Mechanism

A significant shortcoming in the original system was inadequate dynamic response from the voice coil actuator³. Therefore, a flexure mechanism, Figure 3, was designed to introduce stiffness to the voice coil stage. The design combines two identical planar flexures which are flipped 180° about their central axis and pinned together using light press fit dowels. The resulting symmetry ensures a constant spring rate in either direction of rotation of the voice coil. Dowel locations are offset 20° from the mounting holes and provide symmetrical preloads in the direction of coiling to ensure that both flexures experience only coiling loads for any rotation of the voice coil stator within its

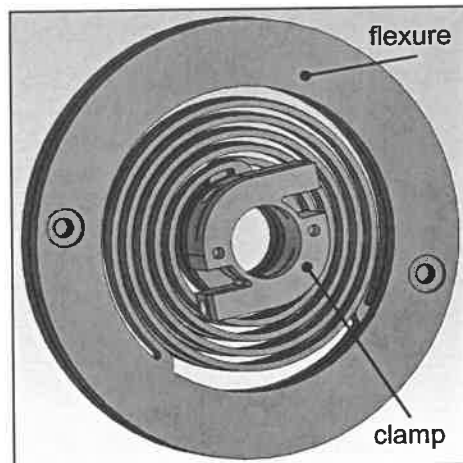


FIGURE 3. Preload flexure assembly.

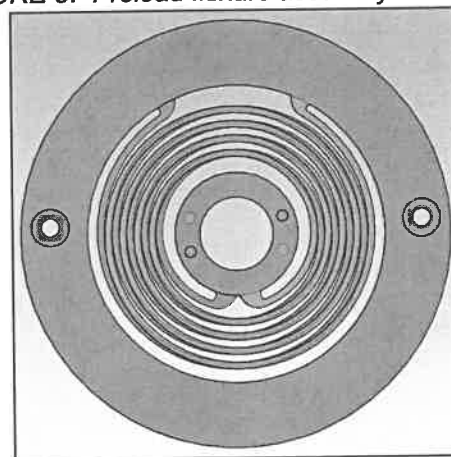


FIGURE 4. Flexure stack showing 180° flipped orientations and 20° offset dowel locations.

$\pm 15^\circ$ range of motion. The assembly fits between the stage base and voice coil rotor, and utilizes a clamp for attachment to the rotating shaft. The flexures are made from 304 stainless to facilitate fabrication via wire-EDM machining.

The flexures were designed to generate an open loop system resonance of 10Hz . While well below the original specification of 100Hz , 10Hz represents the maximum that can be achieved for a $\pm 15^\circ$ range of motion with the torque capabilities of the BEI-Kimco RA29-11-002A rotary voice coil. Based on a $1.4 \times 10^{-5} \text{kgm}^2$ mass moment of inertia for the voice coil's moving mass, the subsequent torsional stiffness, K_t , for a single flexure was $0.028\text{N}\cdot\text{m}/\text{rad}$. The thickness of the flexure, b , was arbitrarily chosen to be 2mm , while a maximum allowable stress, σ_{max} , of 162MPa was selected to ensure a fatigue life greater than 10^8 cycles. Based on closed form design equations for a spiral spring⁴, the corresponding strut width, w , was calculated

to be 0.86mm from

$$w = \sqrt{\frac{6 \cdot K_t \cdot \theta_{max}}{b \cdot \sigma_{max}}}$$

where θ_{max} is the maximum angle of rotation in radians. The 356mm length of the spring section, l , was obtained from the relationship

$$l = \frac{\pi \cdot E \cdot b \cdot w^3}{6 \cdot K_t}$$

where E is the material elastic modulus. A start diameter, D_i , of 12.7mm was chosen to accommodate the diameter of the voice coil shaft. The subsequent 24.16mm finish diameter, D_o , was determined by

$$D_o = \frac{2 \cdot l}{\pi \left(\frac{\sqrt{D_i^2 + 1.27 \cdot l \cdot w} - D_i}{2 \cdot w} - \frac{\theta}{2 \cdot \pi} \right)} - D_i$$

Finally, the 4.96 number of turns, n , was found geometrically based on the length, initial diameter and final diameter.

Once the flexure design was developed analytically, FEA analyses were performed for validation and further investigation. A torsional stiffness of 0.033N·m/rad was predicted from the FEA for a single flexure and was within 16% of the closed-form solution. A maximum stress of 172MPa for a flexure rotation of 35° was then calculated by FEA and was within 6% of the original design target. Finally, the first resonance of the flexure was estimated via FEA to be 160Hz which was deemed satisfactory for system operation.

Closed Loop Control

A second area of effort for improving the dynamic response of the voice coil stage was to implement closed loop control of the voice coil actuator. Figure 5 shows the transfer function obtained without the preload flexure using a PID control algorithm with proportional, derivative and integral gains of 0.07, 1.0 and 1×10^{-6} . Control was implemented using a 2kHz update rate, and achieved a -3dB crossover at 35Hz with a damping ratio of 0.75. 1.33° of overshoot and a 40msec settling time were observed for a step input, Figure 6; while a 0.35° amplitude error and a 52.3° phase error were observed for a 30Hz, $\pm 15^\circ$ sine wave input, Figure 7. A

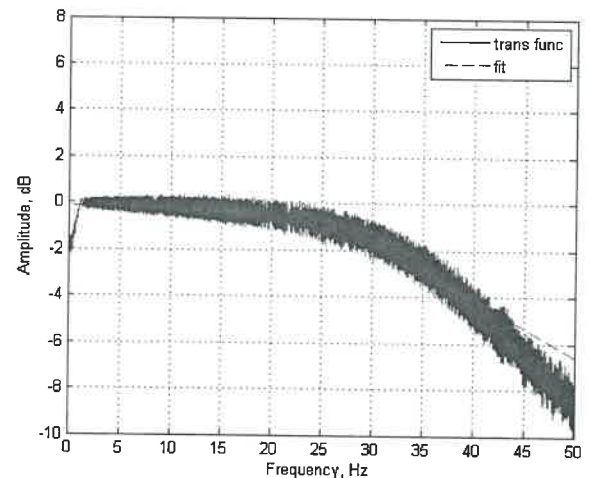


FIGURE 5. Voice coil actuator PID closed loop transfer function.

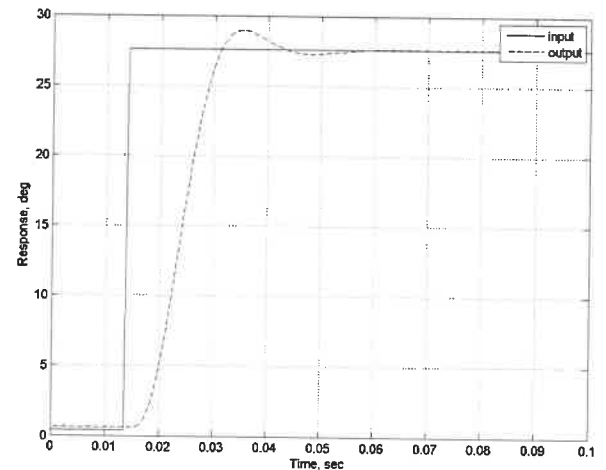


FIGURE 6. Voice coil actuator response to a 27° step input.

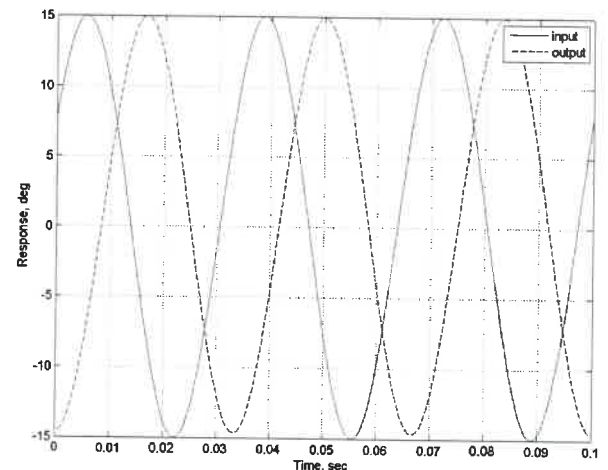


FIGURE 7. Voice coil actuator response to a 30Hz, $\pm 15^\circ$ sine wave input.

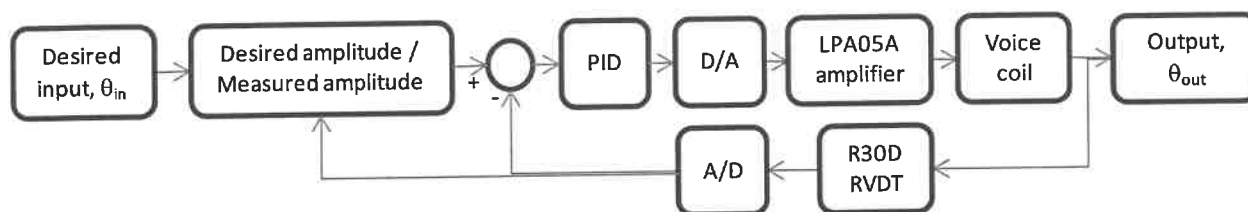


FIGURE 8. PID control diagram with amplitude control.

modified PID control scheme, Figure 8, was utilized to reduce sine wave errors whereby the control signal amplitude was modified based on the desired sine wave response. As a result, sine wave response up to 30Hz consisted predominantly of phase errors with amplitude errors remaining below 0.4° .

SPRING TESTING

Initial spring tests have been performed as Figure 9 shows spring data using a 82.5° initial offset superimposed with a 1Hz, $\pm 12.5^\circ$ amplitude sine wave. Torque values are within the tolerance range of the tested spring design, an early indication of system success. Initial insight into dynamic spring response has also been achieved as the internal friction between the coil windings has been observed to contribute 5-10% of the torque required to rotate the torsion springs. This internal friction is quantified by the step change in Figure 9 that occurs at the extremes of spring travel.

CONCLUSIONS AND CONTINUED WORK

The development of a prototype system for the testing of meso-scale torsion springs continues. Improvements regarding the torque sensor, a flexure mechanism and closed loop control have been implemented as initial spring data has been collected. Continued work is focused on further system improvements, understanding its performance and quantifying uncertainties in spring test measurements.

ACKNOWLEDGEMENTS

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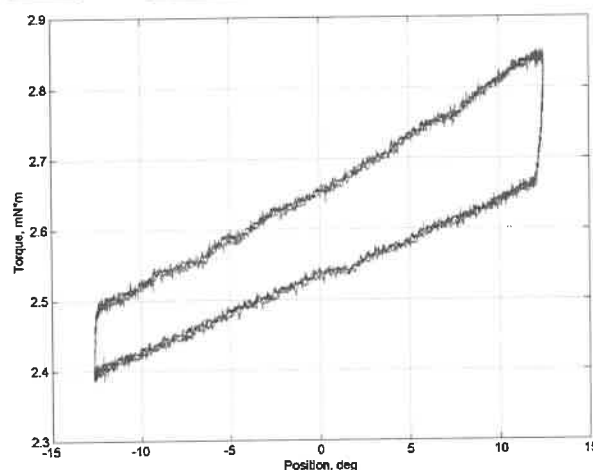


FIGURE 9. Initial torsion spring test data.

has been reviewed and approved for unclassified, unlimited release under SAND2011-5735C.

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A Precision Thermal Press for Polymer Processing

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ABSTRACT

A desktop-size precision thermal press has been developed for polymer processing, targeting micro-fluidic device manufacturing. A three pillars support design is chosen for rigidity and a cruciform flexural joint is used in the base support to achieve passive coplanar alignment. A leadscrew mechanism, together with a stepper motor, is used to generate and maintain the press force. Thermo-electric-devices are used to actively manipulate thermal flow in the system. The developed system is able to control temperature, position and force, precisely.

1. BACKGROUND

Thermal press equipment is not new and there are many such equipment available in the market today [1, 2]. They can be used for the hot embossing process. The hot embossing process begins with the heating of the platens that houses the mold and the substrate. Once the embossing temperature is reached, which is above the glass transition temperature of the substrate material but below its melting point, a compressive force is applied and held for a period of time. This is to allow the polymer to flow into the cavities of the mold to reproduce the features on the mold. Temperature is then lowered below the glass transition temperature while maintaining the applied force so that the embossed substrate will retain these features. Heating in these equipments is generally achieved using cartridge heaters while cooling is through fluid flow with the aid of a chiller. Most commercially available machines are used for handling a wider range of process with an embossing area up to 6 inch in diameter [3].

However, for research and development, these equipments are bulky, lack precision and definitely not energy efficient. This is especially so when used for manufacturing smaller size polymeric products, like the micro-fluidic chips

[4, 5]. As such, a desktop-size thermal press has been developed, which is user-friendly, energy efficient and has high precision. Its structural design, position-force and thermal control will be first discussed. Results of embossing using this thermal press and the needs for future work will be presented.

2. STRUCTURAL DESIGN

One of the key issues to be addressed in thermal press equipment is structural rigidity and to protect sensitive components within the system. Considering the defined specifications as listed below, a 3-pillars structural design was chosen over a 4-pillars for the ease of calibration and over 2 pillars for the rigidity. The working prototype of the thermal press is shown in Figure 1.

- Press Force: up to 2kN
- Coplanar Alignment: Passive
- Operating Temperature: <150°C

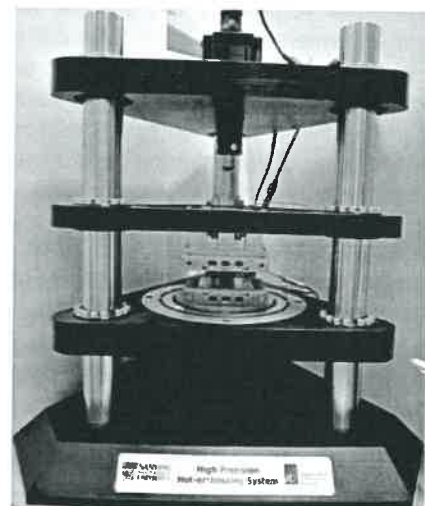


Figure 1: Developed precision thermal press (600mm x 600mm x 600mm)

The open structure of the system allows easy access. "Mecha-lock" is used because of its ease for re-adjustments and its ability to align itself in a concentric manner with the support shafts. Thus, the adjustable frame allows for ease of addition or modification of other required components.

Thermal electric devices, sandwiched by ceramic materials, are brittle and cannot withstand impact or high static loads. As such, a structural frame, with finite element analysis (FEA) using Ansys, has been designed to protect these sensitive thermal electric devices from experiencing directly the applied load. This "load frame" provides an alternative load path, diverting the load away from the components where they are housed, see Figure 2.

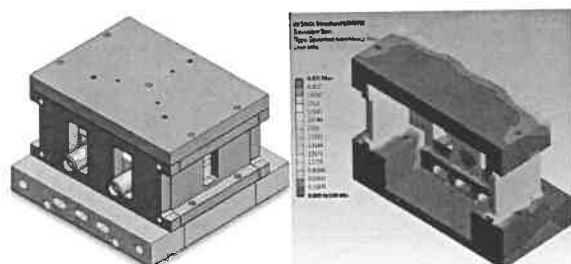


Figure 2: "Load Frame" (left), FEA stress analysis of load frame (right)

As heat is introduced into the system, adequate heat insulation is required to prevent heat from transferring to the other components within the system. This insulation also ensures the safety of the user. A glass fiber resin is used as insulation material because it can withstand temperatures of greater than 200°C with minimal distortion under the required working conditions of up to 2kN press force.

Coplanar alignment is a consideration for the production of good quality parts. A passive design is preferred to eradicate the need for control [6]. A flexure design has been chosen to achieve the coplanarism because it replaces the need for motion stages and control. A cruciform flexural-based joint, as shown in Figure 3 is adopted and implemented on the platen base so that it can flex and provide the necessary self-compensation but maintaining high stiffness in the press direction. This ensures parallelism between the upper and the lower platens. The rotational stiffness of a cruciform flexure joint can be described as follow [6]:

$$K_{\theta z M z} = 2 \left(1 - \frac{0.582t}{d} \right) \frac{Gt^3}{3L} - 0.418 \frac{Gt^4}{3L} \quad (1)$$

$$K_{\theta x M x} = \frac{E(dt^3 + td^3 - t^4)}{12L} = \frac{Gt^4(1+\nu)(\frac{d}{t} + \frac{d^3}{t^3} - 1)}{6L} \quad (2)$$

where G and E are the modulus of rigidity and the Young's modulus respectively; L , t and d are respectively the length, the thickness and the width of the beam.

The positioning of the cruciform features will determine its ability to compensate the tilt effects. The design of the flexure is based on the similar principle of the rotating bearings in a gyroscope. This gives a rotation compliance about the X-Y plane and yet having minimal deflection about the Z axis. The flexure configuration and its mounting ring are machined from a single material, making it a monolithic flexure, and hence reducing the parasitic errors.

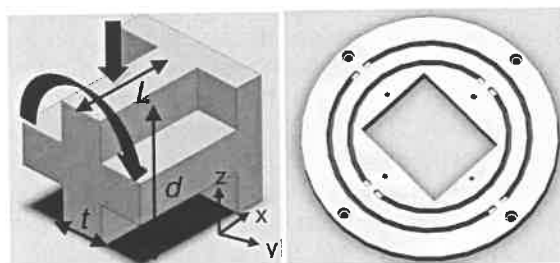


Figure 3: Schematic of a single cruciform flexure (left), platen with a combination of cruciform flexures (right)

Finite element analysis simulation using ANSYS was conducted to verify its performance before this component was fabricated.

3. THERMAL CONTROL

Polymer processing usually involves both heating and cooling. Thermal-electric-devices (TEDs), from Laird Technologies (HT8-12-F2-4040-TA-W6) serve like thermal pumps. They control thermal flow depending on the direction of electrical current flow. The effective cooling of a TED is governed by [7]:

$$Q_{rc} = \alpha IT - K_T \Delta T - \frac{I^2 R}{2} \quad (3)$$

where α is the Seebeck coefficient; I , T , K_T and R are the input current, the temperature, the total thermal conductivity of the semiconductors and the electrical resistance respectively.

Although TEDs serve as a thermal pump, a thermal source and a thermal sink are required during the heating cycle and cooling cycle respectively. Thus, a thermal module has been carefully designed both to house the TEDs and to serve as a thermal exchange itself, as shown

in Figure 4. The fluid channels inside the module are to provide sufficient surface area for exchange of thermal energy, depending on which part of the thermal cycle it is in.

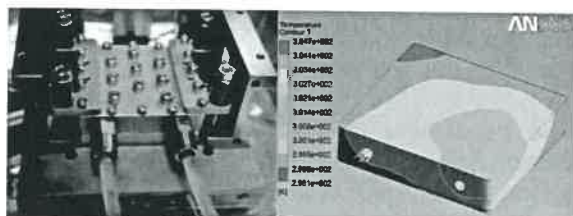


Figure 4: Thermal module (left), CFX temperature simulation (right)

The fluid media used in the channels is water mixed with coolant (Propylene Glycol). This water coolant mixture allows for more heat to be absorbed and transported away during the cooling cycle. The re-circulative liquid is stored in a reservoir, shown in Figure 5. It is incorporated for thermal storage purpose so that the excess heat rejected from the platen during the cooling cycle can be used for the next heating cycle. This helps to ensure that the fluid has sufficient capacity to remove heat during the subsequent cooling cycle. With thermocouples as temperature feedback to the control system, thermal tracking error of the thermal platens is kept well below 0.5°C . A thermocouple was placed inside the fluid reservoir to monitor the water temperature. After running for 10 cycles, each of 10mins, temperature in the reservoir had reached a steady temperature of about 40°C indicating that there was no thermal built-up issue.



Figure 5: Water reservoir

4. POSITION & FORCE CONTROL

A leadscrew mechanism is one of the best options for energy efficient press application. Zaber Linear actuator NA34C60, capable of pressing 2.2kN, with a step resolution of 0.2 microns is used. Kistler loadcell sensor, 9001A, provides force reading up to 7.5kN with resolution as high as 0.1%. PID control architecture is used and force tracking error of $<20\text{N}$ has been achieved. During the embossing process, the upper platen moves until it makes

contact with the substrate sitting on the lower platen. After which, the PID controller takes over for fine adjustment. The control system is based on force control hence deflection experienced by the platen is compensated for. A control flow chart, as shown in Figure 6, depicts the force control in the system.

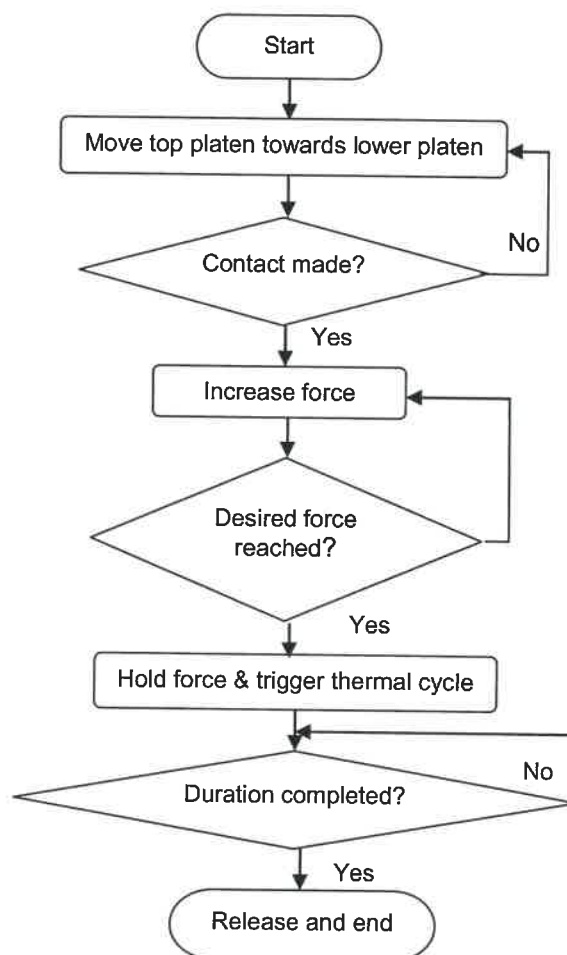


Figure 6: Force control flow chart

5. DATA ACQUISITION, CONTROL & GUI

NI cDAQ is used for online data acquisition and control. Control strategy has been implemented in Labview where friendly graphical user interface (GUI) has been implemented. Recipes for various press applications can be easily entered into the interface. The user's recipe is generated and displayed to ensure that it is the required profile. Once the user confirms the recipe, the cycle can be triggered with the press of a button. Visual displays, such as charts and numerical displays, are used for monitoring purposes, see Figure 7. Although our system has a limited maximum press force of 2kN as compared to Specac, which is able to provide a 50kN force, the advantages of our machine are the precision control of the process parameters, i.e. position, force and temperature.

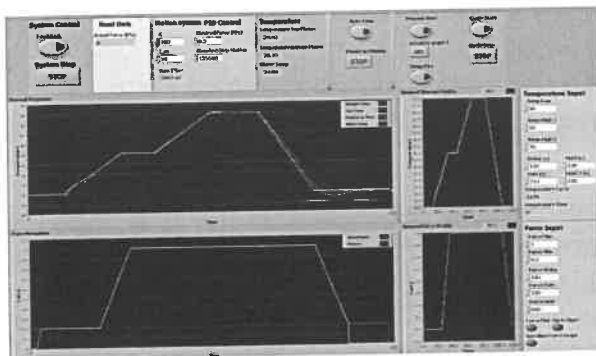


Figure 7: Graphic User interface of precision thermal press system

6. EMBOSSED RESULTS

Protruding features, with dimensions of $2000\mu\text{m}$ (length) $\times$ $200\mu\text{m}$ (breadth) $\times$ $20\mu\text{m}$ (height), were created on a silicon die to serve as a embossing mold. The features were then embossed onto PMMA sheet of 1mm thickness. The experiment was conducted at ambient condition. The total process cycle was 8mins, with a press force of 1.5kN and at a temperature of 120°C . The embossed features on both the mold and substrate are shown in Figure 8. A profilometer was used to measure the features. Figure 9 shows the measured results. The maximum error was found to be $<1.6\%$. The reason for the difference in dimensions between the mold and substrate could be due to the difference in the thermal properties of the mold and the substrate.

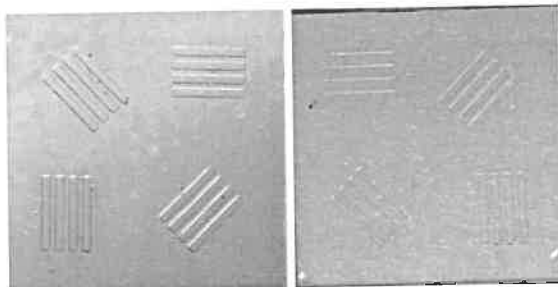


Figure 8: Protruding features on silicon die (left) & Embossed features on PMMA (right)

7. SUMMARY & FUTURE WORKS

A precision desktop thermal press has been developed. The modular system design allows for ease of adjustments and maintenance. Modifications can be easily achieved to suit the needs required by the user. The use of TEDs coupled with a liquid reservoir has shown to be sufficient for the required thermal cycling process. The GUI ensures process parameters can be easily entered and monitored. The control system ensures the precise control of the process parameters. The embossed features has shown 98% accuracy in reproducing the

mold features. The slight deviations can be attributed to the different thermal shrinkage of the polymer and the mold.

Future work for this project is to develop an online estimation of the embossed depth into the polymer substrate. Currently, the displacement information is obtained from the motor controller, which may not be the actual embossed depth because of other material deformations during pressing. In addition, the energy efficiency of the TEDs will be studied to provide a better understanding of their performance in terms of thermal energy recycling.

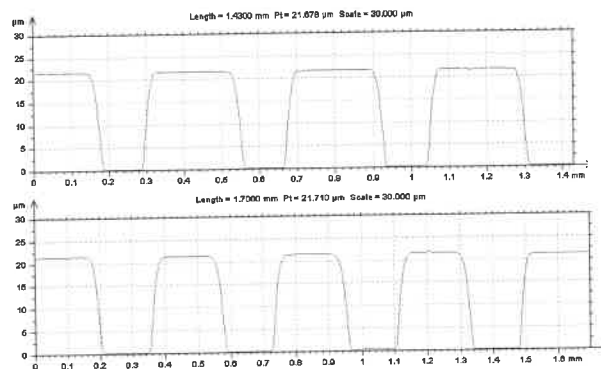


Figure 9: Master Mold Features (Top) & Embossed features on PMMA (Bottom)

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DESIGN AND CONTROL OF A PLANAR 3-DOF PARALLEL MICRO-MANIPULATOR

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INTRODUCTION

A major current focus in mobile electronics manufacturing is making the products such as smart phone, MP3 player and PMP smaller in size due to portablization and miniaturization. Traditionally, these small size products are produced by equipment, which demands large rooms, large amount of energy and high labor cost. Micro-factory, however, can minimize above problems by downsizing the manufacturing equipments, reducing the working space and lowing energy consumption. For these reasons, micro-factory increasingly obtains public attention [1].

In order to improve the accuracy and productivity of the micro-factory process, precision aligning operation is one of important processes. Many researchers and industries have studied parallel manipulators, which have good level of accuracy and precision. Compared with serial manipulators, parallel mechanisms are capable of high precision, light weight, high stiffness, substantial payload, stable operation and free arrangement of actuator. However, they present limited workspace and inevitable singularities [2].

For the reasons, much research in recent years has focused on the optimization problem and control of parallel mechanisms to have wide workspace and improve the mechanisms' performance [3, 4]

This paper presents the design and simple closed-loop control of a planar three degree-of-freedom (DOF) parallel manipulator. In the design process, inverse kinematics and optimization problem of the manipulator is presented. The manipulator adopts PRR (Prismatic-Revolute-Revolute) mechanism.

Experimental tests have been carried out with high resolution capacitive sensors in order to verify the performance of the mechanism.

POSITION KINEMATICS

The manipulator presented in this paper has some required specifications. Those are as follows; (1) dimension: $100 \times 100 \text{ mm}^2$, (2) moving rage of end-effectors: $x, y \sim \pm 1 \text{ mm}$ and $\theta_z \sim \pm 0.5^\circ$. Position and direction of the actuators which are arrayed 120° apart symmetrically are configured to satisfy the requirements.

Figure 1 shows a schematic diagram of the 3-PRR manipulator. Motion of the parallel manipulator is analyzed using the solutions of inverse and forward kinematics [5]. From the geometry of kinematic chains connecting the base to the platform, the coordinates of point C_i can be written as,

$${}^B C_i = {}^B P_H + {}^B R_H {}^H C_i = {}^B A_i + \rho_i e^{j\theta_i} + b_i e^{j\psi_i}, i=1, 2, 3 \quad (1)$$

where B is a reference frame on the base, H is a reference frame on moving platform, ${}^B C_i$ is C_i 's position with regard to B , ${}^B P_i$ is P_i 's position with regard to B , ${}^H C_i$ is C_i 's position with regard to H and ${}^B R_H$ is rotation transformation from H to B . This finally leads to a quadratic equation in ρ_i which can be solved and leads to

$$\rho_i = M_i \pm N_i, i=1, 2, 3 \quad (2)$$

where

$$M_i = ({}^B x_{C_i} + {}^B x_{A_i}) \cos \phi_i + ({}^B y_{C_i} - {}^B y_{A_i}) \sin \phi_i \quad (3)$$

$$N_i = \sqrt{b_i^2 - S_i^2} \quad (4)$$

$$S_i = ({}^B x_{Ci} - {}^B x_{Ai}) \sin \phi_i - ({}^B y_{Ci} - {}^B y_{Ai}) \cos \phi_i \quad (5)$$

${}^B x_{Ci}$ and ${}^B y_{Ci}$ are constant parameters describing the geometry of the platform because they are the coordinates of point C_i in the coordinate frame (H) moving with the platform.

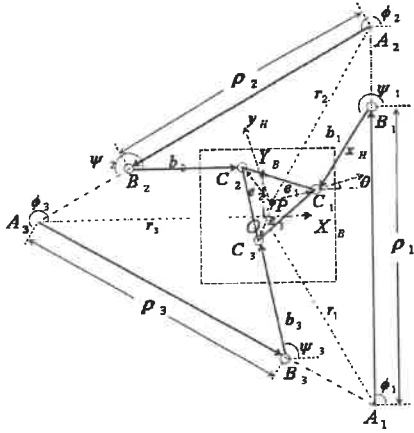


Figure 1. Schematic diagram of a 3-PRR manipulator and possible workspace. B is fixed on base and H is moving platform.

This provides a closed-form solution of the inverse kinematic problem. From Eq.(2), displacement of the prismatic joints (ρ_i) can be computed from the position (${}^B x_p, {}^B y_p$) and orientation (${}^B \theta$) of P . The several solutions of position kinematics have been implemented in MATLABTM. Eq.(2) has been also used to compute the displacement of the prismatic joint actuators in the closed-loop control.

DESIGN OPTIMIZATION

OBJECTIVE OF OPTIMIZATION

Workspace of the parallel manipulator is the one of the most important performances due to its characteristics. The design optimization aims to determine the optimum geometric parameters of the parallel manipulator having maximum workspace satisfying required constraints. In this paper, therefore, the design optimization of the parallel manipulator is based on the formulation of workspace maximization and proper constraints.

To calculate the objective function and check the workspace, the entire possible workspace is introduced by a dashed square as shown Figure 1. A numerical solution can be obtained with following methods [6].

- Step1: Consider the entire possible workspace of the manipulator.
- Step2: Select a large number of points (n_{total}) which are individually within the predefined square.
- Step3: Determine if ρ_i falls within manipulator workspace by solving the inverse kinematic problem. This is possible by checking if ρ_i is between ρ_{min} and ρ_{max} or not.
- Step4: Count the number of points that fall within the workspace, n_{in} .
- Step5: From objective equation, Eq.(6), objective function can be calculated.

$$W = \frac{n_{in}}{n_{total}} \quad (6)$$

CONSTRAINTS

The proposed manipulator in this paper will be applied to precision positioning devices in a micro-factory that requires compact size. According to the requirements, constraints for the design optimization are given as follows.

- The prismatic joint are fixed 120° apart symmetrically on the base.
- For the symmetrical condition, let $b_1 = b_2 = b_3 = b$, $e_1 = e_2 = e_3 = e$, $r_1 = r_2 = r_3 = r$.
- In order to avoid intervention between prismatic joint, the lower and upper bounds of the prismatic lengths are defied.

$$0 \leq \rho_i \leq \sqrt{3}r \quad (7)$$

- The last constraint is related to the mechanism assembly.

$$b + e = r/2 \quad (8)$$

DESIGN VARIABLES

The geometric parameters (e, r, b) are considered as design variable. Hence, the design parameters x is given as follows.

$$x = [e \ r \ b]^T \quad (9)$$

RESULT OF DESIGN OPTIMIZATION

The design optimization for a planar 3-PRR parallel manipulator can be solved by finding the optimum design parameters x of a planar 3-PRR manipulator in order to maximize its workspace.

The problem can be written as following Eq. (10).

$$\begin{aligned}
 & \text{maximize} \quad f(x)=W \\
 & \text{over} \quad x=[e \ r \ b]^T \\
 & \text{Subject to : } g_1: 0 \leq \rho_i \leq \sqrt{3}r \\
 & \quad \quad \quad g_1: b+e=r/2 \\
 & \quad \quad \quad x_{lb} \leq x \leq x_{ub}
 \end{aligned} \tag{10}$$

where x_{lb} and x_{ub} are the lower and upper bounds of x , respectively.

TABLE 1. Bounds of the design variables.

Design variable	e	r	b
x_{lb} [mm]	0	59	35
x_{ub} [mm]	10	61	45

TABLE 2. Optimization results.

Objective	Design variable		
W (Workspace)	e [mm]	r [mm]	b [mm]
0.2608	9	59	41

MATLAB™ Optimization Toolbox was used to solve the Eq.(14). Table 1 shows lower and upper bounds of the design variable. Optimal solution of the problem is given in Table 2. As a result of these tables, the workspace of the proposed 3-PRR manipulator is maximum value as 0.2608. Also, when e is 8.2 mm, r is 56 mm and b is 39 mm, the workspace analysis was carried out to compare with the optimum value. This result was 0.1429 and smaller than optimum value.

EXPERIMENTS AND RESULT

From the design optimization, the prototype of a planar 3-PRR parallel manipulator was fabricated as shown in Figure 2. The size of proposed 3-PRR manipulator is 100 x 100 x 65 mm³. Three 21000 series size 8 step motors (Haydon kerk™) actuate prismatic joints and the tetragonal plate are connected to the moving platform in order to make the capacitive sensors possible to measure motion of the moving platform.

The experiments are carried out for performance test. The photo of the experimental setup is shown in Figure 3. The prototype has inevitable tolerance and this is overcome by using closed-loop control, which can also allow compensation

for other error motion. As shown in Figure 4, in the closed-loop control, input x , y and θ is calculated how much displacement the step motors should move by inverse kinematics, which is possible by inputting the number of pulse signal to step motors. To measure motions of the 3-PRR parallel manipulator and compensate of the error motion, three Microsense II 5810 capacitive sensors (ADE Technologies) were used. Using the output data from the three sensors, the actual x , y and θ displacement is calculated again and the error is also calculated. For the real time closed-loop control, DS2002/DS2003 A/D, DS2102 D/A and DS1005 PPC boards (dSPACE) are used.

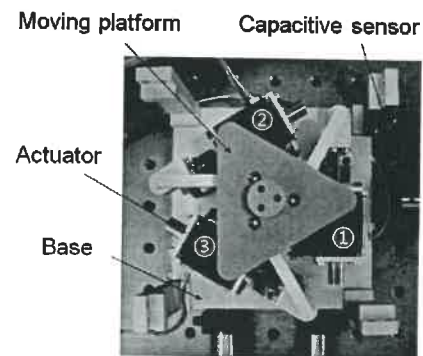


Figure 2. Photo of the proposed manipulator

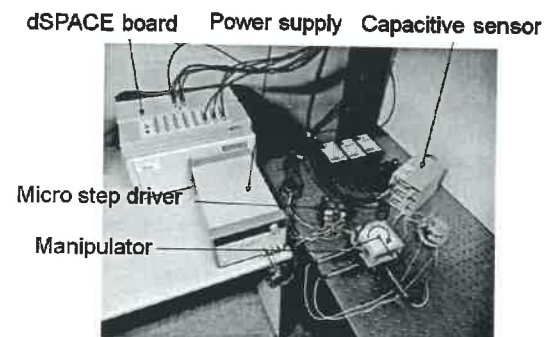


Figure 3. Photo of the experimental setup

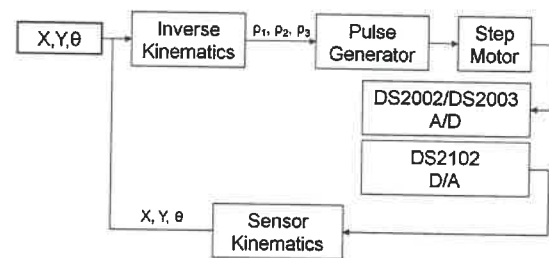
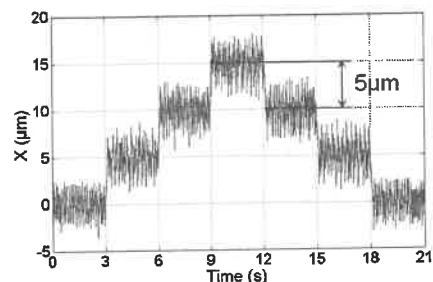
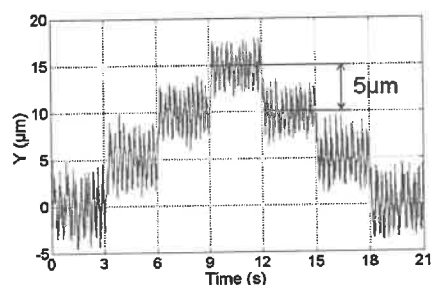


Figure 4. Block diagram of the closed-loop control

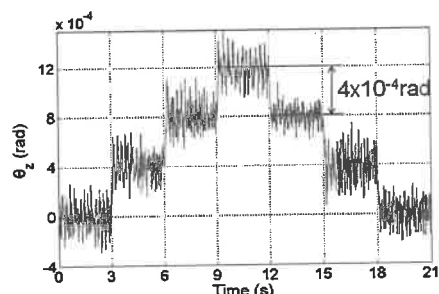
Figure 5 shows the results of closed-loop resolution tests. As the results, resolutions were $\pm 2.5 \mu\text{m}$, $\pm 2.5 \mu\text{m}$ and $\pm 0.2 \text{ mrad}$ for X-axis translational motion, Y-axis translational motion and Z-axis rotational motion, respectively.



(a) Translational motion in X-direction



(b) Translational motion in Y-direction



(c) Rotation motion in θ -direction

Figure 5. Closed-loop resolution test results.

CONCLUSION

In this paper, conceptual design of the planar 3-PRR parallel manipulators for micro-factory was carried out with the design optimization. Using position kinematics and design tool, the optimum values having maximum workspace was obtained.

And then, a planar 3-PRR parallel manipulator was fabricated and the inverse kinematics of the mechanism was verified through real-time closed-loop control. Through the closed-loop control, the error motion of the 3-PRR

manipulator was reduced. As the results, the resolution of the 3-PRR manipulator reaches $\pm 2.5 \mu\text{m}$, $\pm 2.5 \mu\text{m}$ and $\pm 0.2 \text{ mrad}$ for X, Y and θ_z direction, respectively.

In the future work, experiments for the workspace verification will be carried out.

ACKNOWLEDGEMENT

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FORM MEASUREMENTS OF SPECULAR PARTS BY COMBINING BEAM PROPAGATION, OPTICAL SCATTERING, AND PHOTOGRAMMETRY

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INTRODUCTION

Spherical optics produced by traditional polishing methods are easy and inexpensive to manufacture. However, the use of these types of optics is currently limited by aberrations induced by the spherical geometry of the component. In order to alleviate these aberrations, and at the same time minimize the number of elements used in an optical system, aspherical or free-form optical elements are preferred. While high precision, multi-axis diamond turning machines are capable of manufacturing the required forms, thermal environmental changes while machining, especially large parts several meters in size, can lead to significant form deviation. Final inspection is not typically completed while the part is on the machine; rather the part is measured externally with a coordinate measuring machine (CMM), an optical scanner, or an interferometer. Shortcomings in these techniques include: contact measurements can compromise the surface quality, interferometry is best suited to nearly flat or spherical components, and common optical scanning techniques often do not meet desired measurement uncertainties. Further, removing the part for the measurement breaks the connection between the part and machine-tool coordinate systems making re-machining difficult.

Currently, the limiting factor in part manufacturing is the precision metrology used to assess surface form errors. However, if measurements of the form could be performed in-situ, while the part is being machined, then better parts could be manufactured where inaccuracies in the surface form caused by the machine tool can be corrected. In this paper, a new measurement system is proposed that combines photogrammetry and optical scattering with specular reflection of a laser beam off the workpiece surface [1]. A laser beam is directed towards and reflected from the surface to be measured, passing through two windows in

transit. Low levels of scattering occur when the beam passes through the windows and the beam location at the window becomes observable. Photogrammetry is used to determine the coordinates of the beam locations on the two windows, which is then used to determine the incident and reflected beam orientations. Vector calculus is then applied to determine the surface normal, and finally the profile is extracted via integration. For this metrology technique, we suggest that uncertainties in the angle of the incident beam with respect to the surface normal, directly correlates to the uncertainties in the local slope and the sagitta on the part at a particular point. Simulating random Gaussian noise in the slope, or incident angle, associated with a simple parabolic form on the orders of 2:10000 and 3:100000, produces an overall sagitta deviation error of approximately 200 nm and 10 nm, respectively. Photogrammetry can realize angle uncertainty levels as low as parts in 10^4 and parts in 10^5 in some cases, suggesting this technique can realize form uncertainty at the 10-100's nm level. For certain components such as ogives, this measurement uncertainty is adequate, particularly when it can be done in-situ. Further, the method proposed here, is cost effective, non-contact, and has a large dynamic range capable of measuring arbitrary free-form and aspheric surfaces.. Though the metrology system has been shown to be a promising method of in-situ testing, there are several factors related to the setup that were investigated in order to further improve the system. First, a study of window medium – previously a limiting factor for system automation due to unwanted secondary reflections – is discussed. Second, the fully automated system incorporating image processing, photogrammetry processing, laser positioning, and profile calculation is described which leads to high measurement resolution and a shorter measurement time.

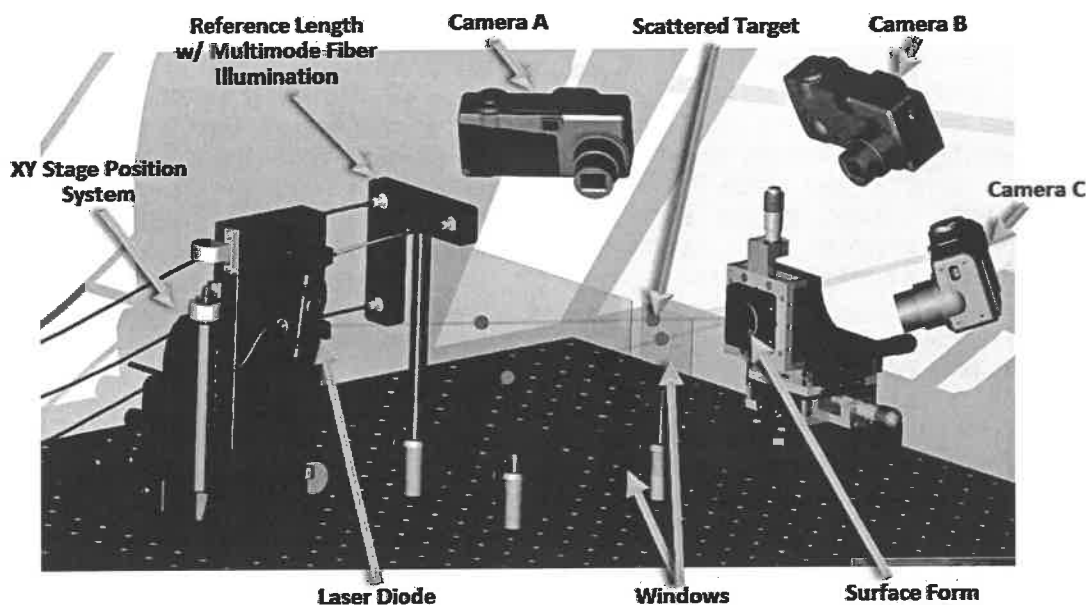


FIGURE 1. In-situ metrology setup. The locations where the incident and reflected beams pass through the tinted windows are determined by photogrammetry, using the photographs taken by three cameras. These spot coordinates determine beam locations and orientations, and therefore the surface slope on the part can be measured.

SYSTEM SETUP AND ADVANCEMENTS

A detailed description of the measurement system is given in reference 1, and is shown in FIGURE 1. Previously process automation was limited by multiple specular reflections between window-to-window surfaces and window-to-part surfaces. This made automatic spot identification difficult. This occurred when using windows that were lightly roughened and tinted to aid in diffusely scattering the laser light on the window surface [1]. With the use of anti-reflection (AR) windows, this issue has been significantly reduced where the reflectivity is less than 1% for the laser wavelength of 650 nm. Eliminating unwanted specular reflections from the windows eliminates unwanted laser spot profiles within the area of the windows. The resulting image of the setup can be seen in FIGURE 2, corresponding to Camera A in FIGURE 1. Because only the spots of interest are visible from the cameras now, the ability to fully automate the system is realized. The measurement efficiency and resolution was limited by the rail used to raster the laser across the surface. In the initial experiments the rail was operated manually and had 1 μm resolution in one axis. It was replaced by a 2-axis, motorized actuation position system with 50 nm and 7 nm resolution in the x and y axes, respectively. This increases both measurement density and speed.

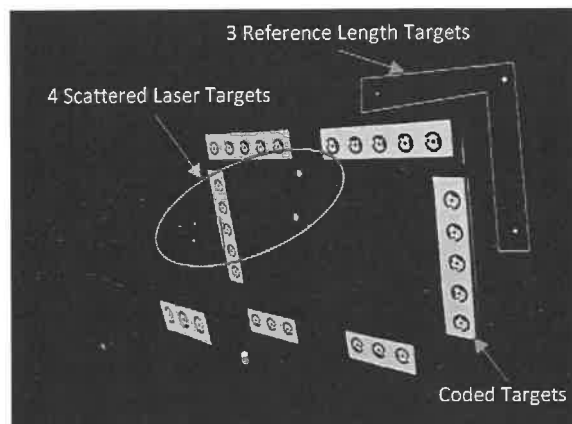


FIGURE 2. Measurement setup showing the four scattered laser dot targets, three reference length targets and coded targets.

SYSTEM AUTOMATION AND PROCESSING

An algorithm is being developed using MATLAB™ to centrally command remote image capturing, initial image processing, photogrammetric software communication, LabVIEW communication to drive the motorized actuators, and slope profile generation.

Automation is accomplished in MATLAB™ using dynamic data exchange (DDE) and ActiveX. With DDE and ActiveX, MATLAB™ can be used as a client application to initiate and maintain conversations with server applications, such as

Photomodeler® [2], PSRemote [3], and LabVIEW. Using one application to drive all subsystems increases the user-friendliness of the system. Each point had to be manually marked during the initial testing [1]. Although Photomodeler® has the capability to automatically mark targets, it can mark false, or unwanted, targets created by hot spots or secondary beam reflections. This results in unwanted or incorrect point coordinates.

The following outlines the steps involved in processing the photographs:

MATLAB™ communicates with LabVIEW, over an ActiveX server, where a Virtual Instrument (VI) is used to drive the x and y stage actuators via an RS232-to-USB communication port to the desired, pre-determined, raster location. Once the laser has been positioned to the desired xy location, MATLAB™ is used to communicate with PSRemote [3], a remote capture software used to gather photographs and to control camera settings such as shutter speed. Three photographs of the windows are taken simultaneously at their differing orientations. Each image must include the coded targets on the windows, and the reference length. The coded targets are required to assist in automatic image processing. Coded targets are points with a unique pattern that can be automatically detected, and cross referenced in several photographs by Photomodeler® [1,2]. At least six coded targets must be referenced in two or more images for successful processing and bundle adjustment. The reference length is required to scale the dimensions correctly.

The photographs are imported into MATLAB™ for initial image processing. The seven dot targets, four from the scattered laser light generated on the windows and three from the reference length as seen in FIGURE 2, are identified (marked) using MATLAB™ filtering techniques. This helps alleviate the automatic marking of false targets in Photomodeler®. First different color channels are filtered, depending on the points of interest. Currently, the red channel is examined for the red laser dot targets the blue channel is examined for the blue reference length dot targets. Each color channel image is converted to a binary (black and white) image with an intensity threshold adjustment depending on the lighting conditions. White areas of the binary image that are considered too large or too small to be dot targets are

filtered out. Morphometric parameters, such as object roundness and equivalent diameter are used to identify the dot targets of interest. The centroid of the dot targets within a specified region of interest is then calculated. The region of interest is a boundary, usually defined as the area covered by the windows and the reference length, where the dot targets are expected to be found. Finally, the pixel coordinates for the dot targets, are obtained. These co-ordinates, along with the images, are then exported to Photomodeler®.

Photomodeler® automatically marks the coded targets, uses the pixel coordinates obtained by the image processing to mark seven dot targets from the laser spots and reference length, and orients the cameras. Once all images have been marked, Photomodeler® is then directed to automatically reference each correlating point in the images. For example, Point 1 on Image 1 is referenced to Point 1 on Image 2 and Point 1 on Image 3 [1]. A final bundle adjustment process is implemented where the best estimate of the xyz coordinates of the dot targets are determined. This final bundle adjustment increases the accuracy of the measurements by incorporating the fact that points referenced in the three photos represent the same location in space. The measured points can therefore be used in the same way as the coded targets to minimize the measurement residuals in the photogrammetry process. These xyz coordinates are then sent to the profile generation algorithm to calculate the sag or form of the object being measured, as discussed in [1].

TESTING AND RESULTS

A 12 mm aperture spherical convex mirror was measured using the proposed metrology technique. The laser was rastered over the test part in increments of 0.5 mm in the x-direction and 1.0 mm in the y-direction. The profile results can be seen in FIGURE 3. The radius of curvature for a best fit sphere to the profile data is 52.159 mm. A radius bench measurement of the mirror determined the curvature to be 51.922 mm, a 0.5% difference.

FIGURE 4 shows the residuals after removing a fitted sphere from the measurement data. Bias exists in the residual data along the raster lines. This can be caused by several factors, including errors induced by the profile calculation algorithm, errors induced by the misalignment of the test part, errors in the photogrammetry

calculations of the xyz coordinates of a point, or any combination of the above.

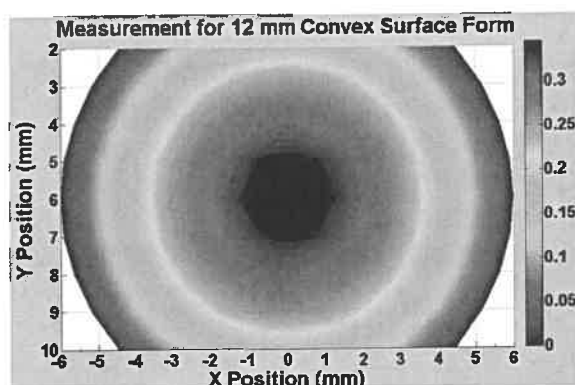


FIGURE 3. Surface form measurement for a 12 mm aperture convex spherical mirror where the R.O.C. is 52.2 mm and the peak-to-valley sagittal is 0.346 mm.

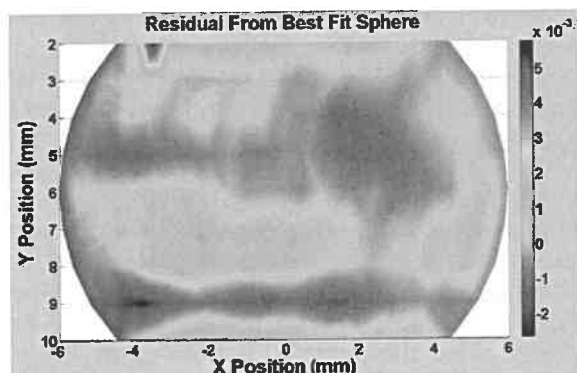


FIGURE 4. Residual after fitting and removing a sphere to the measurement data. The peak-to-valley residual is about 7 μm .

As described in [1], the surface profile calculation algorithm uses only 2-dimensional information about the local surface normal to calculate the profile in individual slices in the y-direction and consequently some 3-D information is lost for the calculation of the surface profile. Misalignment of the test part to the plane of the incident and reflected beams can slightly alter the calculated profile because of the current data analysis technique. A slight shift in the true xy measurement position, compared to the xy raster position, can be present. Some error is induced in the local sag calculation at the edges because it is assumed in the algorithm that each scan starts from the theoretical boundary of the test part. Finally, because the total sagitta (sag) of the profile is gathered by integrating the local sag at each measurement point, error induced by photo-

grammetry, i.e., error in the true xyz position of a laser spot, will also alter the profile.

Ideally the targets identified and referenced in the photographs for the photogrammetry measurement have well defined boundaries. This leads to lower error in the calculation of the xyz coordinate of the dot targets or spot. It has been shown that the standard deviation in spot coordinates is approximately 10-30 μm . This value is sensitive to incident angle. For an angle of incidence of about 4° the uncertainty is still a few parts in 10^3 , however, if the angle of incidence were doubled, the results show an uncertainty of about 2 parts in 10^4 . Due to the irregularity in the surface of the window and laser speckle, the spot contrast is not high enough to further improve results by increasing the angle even more. This has been reported as a reason for avoiding the use of laser dot targets in traditional photogrammetry [4]. Current research is underway to reduce the spatial coherence of the laser beam, thereby reducing speckle and generate semi-transparent windows with uniform scattering properties.

SUMMARY

The measurement system (mechanical and software aspects) described in this paper has been fully automated, and now more complex forms and full aperture measurements can be taken. In addition, it is expected that reducing the spatial coherence of the laser beam will improve measurement results.

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Using an optical software package to analyze photogrammetry system errors

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INSTRUCTIONS

Photogrammetry is a metrology technique that can determine the geometry of an object from several 2D photographic images taken from different locations. Triangulation techniques are applied to the 2D images to generate a 3D, spatially consistent model. Photogrammetry has been widely used for generating 3D geometric models in fields as diverse as architecture, engineering, industry and physiology [1].

It is known that the overall accuracy obtainable with photogrammetry is affected by several inter-related factors, such as camera resolution, size of target object, number of pictures taken, mechanical stability of the camera, and camera locations [2]. Error evaluation of a photogrammetry system is necessary to characterize its uncertainty. Although much work has been conducted to evaluate the errors associated with the cameras used in photogrammetry [3], little has been done to evaluate the uncertainties associated with the whole photogrammetry system (software and hardware) due to its complicated nature. The combined errors for a system depend on specific built-in software algorithms and other operation conditions (e.g. temperature variations) that are generally difficult to determine. An accuracy assessment methodology for digital photogrammetry was reported in [4]. The accuracy was analyzed by means of comparison with known scale-bar length. In particular, the photogrammetric point, distance, and scale accuracy were considered. However, environmental conditions such as temperature are unavoidable and these also influence the accuracy of a photogrammetry measurement. As indicated in Ref. 4, the uncertainties of the scale bar itself, which are induced by its thermal and mechanical properties, must be considered

in the accuracy assessments of photogrammetry.

In this paper we present an alternative approach to evaluate the measurement uncertainty associated with a commercial photogrammetry software package. An advanced optical simulation software package (FRED[®] [5]) can simulate a virtual 3D object and generate virtual photographs of this object via optical scattering and ray-tracing. As the external environmental influence can be eliminated in simulation, this approach, based on the principles of optical propagation, gives us nominally exact photographs of a geometrically defined object. We then use a photogrammetry software package Photomodeler[®] [6] to process these photographs to generate the 3D model of the object. The discrepancy between the known geometry of the object and the dimensions as reported by PhotoModeler[®] are on the order less than 1 part in 10³. Camera calibration is also simulated by this virtual photogrammetry system, which allows us to compare the calibration parameters determined by the photogrammetry analysis to the values as determined by ray-tracing through the virtual camera. These results show that optical simulation packages are an effective method of assessing uncertainty in photogrammetry.

PHOTOGRAMMETRY SYSTEM MODEL

Figure 1(a) illustrates a virtual photogrammetry measurement system which includes a light source, three cameras, and an object (either a box or a calibration grid for the simulations described here). We use FRED[®] and its 3D computer-aided-design environment to simulate all elements, and then use its ray-tracing capabilities to project, scatter and propagate rays through the environment to a virtual camera analysis plane. This allows us to simulate photographs of an object of known dimensions

from first principles. It also allows us to generate photographs by tracing rays through a lens assembly representative of an actual camera. We choose a Cooke triplet (Figure 1(b)) in the camera model, as it has been shown to lead to high quality imaging with minimal optical aberrations [7]. The camera consists of a Cooke triplet lens system, an aperture stop and an image plane. The diameters of the three lenses are 19 cm, 10 cm and 15 cm, respectively. The corresponding effective focal length of this triplet is 49mm. In our simulation, the object to be photographed is placed at the coordinate system origin. Although the photographs of the object are taken by cameras at different location, these locations are chosen so that the distance between this object and each camera is the same. Therefore the image plane of this triplet is determined by imaging a point object at the origin. Here the plane is a 1000×1000 discrete grid of pixels and its size is 10.16 cm×10.16 cm. Thus, the configuration simulates photographs taken with a reasonable quality 1 megapixel digital camera. An aperture stop with inner diameter 2.54 mm is placed between the left and the middle lenses to mitigate aberration effects and improve image quality.

The object is a 20 ×16 ×4 inch box (Figure 2). Several points with specific spatial coordinates are labeled on its surface. The distance between point 1 and point 21 is set as the reference length, which is 12 inches. The reference length is required to scale the object.

As described earlier, ray tracing is used to generate virtual photographs of the system modeled within the FRED® environment. The light source is represented by an array of parallel rays that propagate straight down toward the object. The size of the source array can be varied, and we use an array size of 500 × 500 for our final results. When each ray hits the object surface, a collection of scattered rays is generated based on the defined surface properties of the object. For either the calibration grid or the box, the white surface scatters the incident rays, while black surfaces absorb them. The rays scattered by the object pass through the camera lenses and reach the analysis plane to form an image of the object. Three different light source configurations were investigated to determine which produced the best virtual photograph. They included a collimated planar source, a point source and a Gaussian (laser) source. In all cases the wavelength is 550 nm, a

wavelength which minimizes aberrations in the Cooke triplet. The wavefields emitted by the collimated planar source and the point source are incoherent, while the laser source is coherent. Three photographs of the calibration grid resulting from these different sources are shown in Figure 3. It can be seen that the photograph resulting from the collimated planar source is of the highest quality. This is because the illuminating wave-fronts emitted by other two types of sources are non-uniformly distributed on the object resulting in light intensity gradients across the photographs. Thus we use collimated planar source in the following simulations.

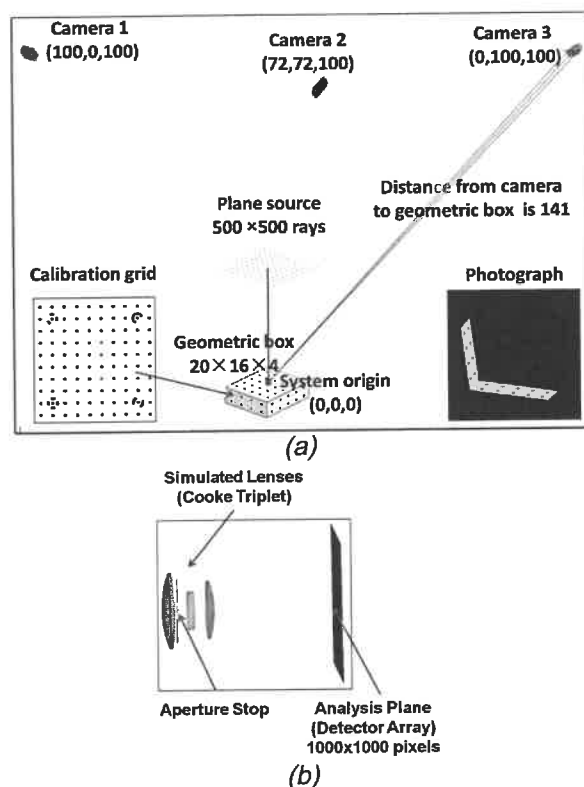


FIGURE 1(a). Illustration of virtual photogrammetry measurement system, (b). Illustration of virtual camera.

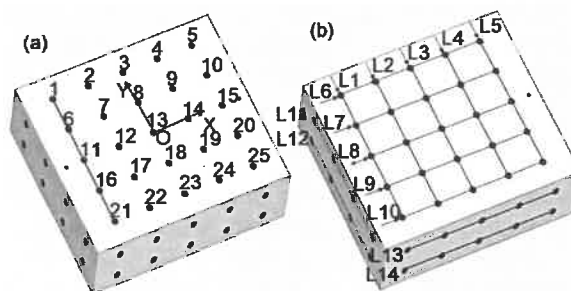


FIGURE 2. (a) The reference length is defined from point 1 to point 21, which is 12 inches. (b)

Fourteen lines (L1-L14) are defined on the surface of the box.

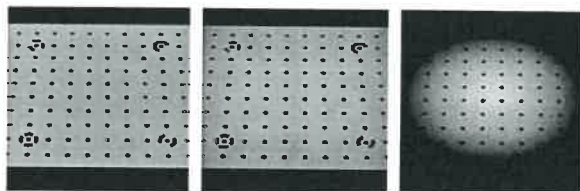


FIGURE 3. Photographs of calibration grid simulated with collimated planar, point, and laser beam illumination sources, respectively (from left to right).

As demonstrated by Figure 3, images of objects can be simulated by virtual cameras in FRED®. However, these images cannot be directly processed by PhotoModeler®. This is because a real digital photograph contains data of the corresponding camera such as the focal length and the image size. These data are needed to create what is called an exchangeable image file (EXIF) and is necessary for photogrammetry processing. Therefore we use the image data obtained in FRED® to generate JPEG image files in Matlab and embed the data of focal length and image size of the virtual camera by using a free software Exif Pilot [8]. Upon performing these steps, the photographs can be processed by PhotoModeler® and the geometric parameters of the reconstructed 3D model can be obtained.

PHOTOGRAMMETRY MEASUREMENT SIMULATION

In general, a photogrammetry measurement contains two steps, camera calibration and object measurement.

In principle, camera calibration is also a photogrammetry measurement where the target object is the calibration grid (Figure 1(a)). The camera calibration step determines the internal parameters of a camera such as focal length, film size, principal point and lens distortion. This calibration enables greater precision when a 3D model of an object is reconstructed from images. To calibrate the camera, several photographs of the calibration grid are taken. The camera is mounted at four different positions, namely front, back, left and right sides of the grid, respectively. At each position, the camera is oriented at three different angles by rotating 0°, 90°, and 270°, respectively. These twelve photographs are processed by PhotoModeler® to determine the virtual camera parameters.

Object measurement is performed by taking photographs of the geometric box from different positions and orientations using calibrated cameras. High quality photogrammetry measurements require photographs taken with large angle variation, good focus, and good exposure. To accommodate this, the camera is mounted at three different positions whose spatial coordinates are indicated in Figure 1. One of the photographs is shown in the inset of Figure 1. With pictures of this box taken from different positions, a 3D model of the object is reconstructed and the coordinates of the originally defined dots are calculated by PhotoModeler®. By comparing the photogrammetry results with the nominal object properties, the errors associated with the entire photogrammetry process can be analyzed.

RESULTS AND ANALYSIS

PhotoModeler® processing of the camera calibration photographs results in estimated values for the focal length and image size. The focal length is 49.8894 mm and the size of the image plane is 10.1596 cm × 10.1596 cm. By comparison with the corresponding parameters originally defined in the previous section, a close correlation between the camera parameters obtained from optical simulation and those obtained through the photogrammetry calibration process is observed.

Figure 4 depicts the measurement errors associated with the 14 different line lengths defined in Figure 2(b). We define the normalized error as the difference between the photogrammetry-determined length and the defined length in the CAD environment divided by the defined length. These normalized errors are at a level of less than 1 part in 10³.

Figure 5 illustrates the spatial distribution of the measured co-ordinates of the 25 points identified in Figure 2(a). It is observed that after reconstruction of the 3D box in PhotoModeler®, these 25 points are no longer on the same plane. The deviation from the reference plane tends to be larger as the point moves further away from the origin. A fitted plane based on the measured positions of 25 points is shown in Figure 6. Its angle with respect to the reference plane $z=0$ is 0.058 degrees.

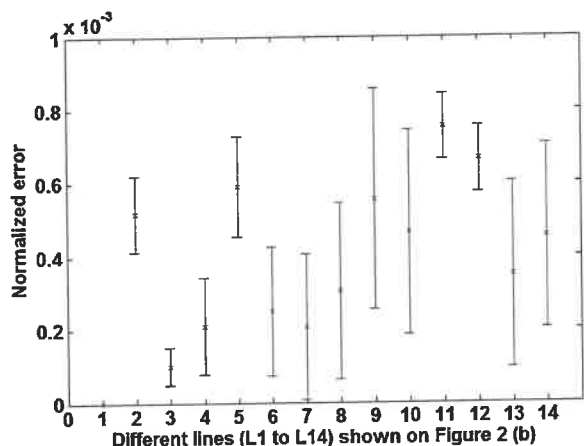


FIGURE 4 Photogrammetry measurement errors of L1 - L14. The cross signs represent the average normalized errors obtained from ten repeated measurements conducted in optical software. Error bars illustrate the corresponding standard deviation of these ten measurements.

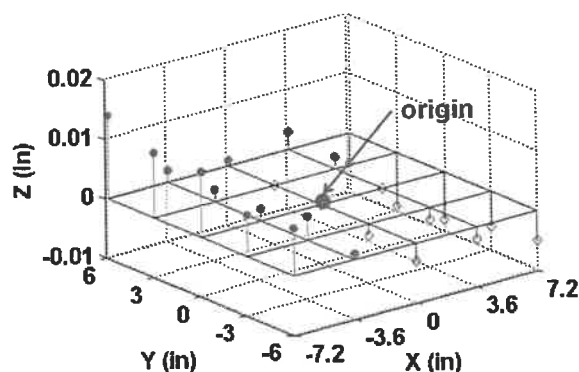


FIGURE 5 Plane $z=0$ is set as the reference plane. The circles represent the points above the reference plane while the diamonds represent the points below the reference plane.

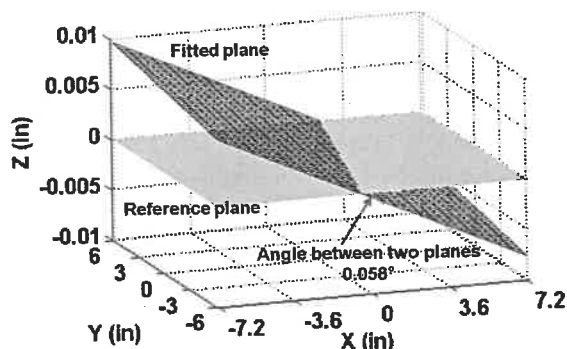


FIGURE 6 Angle between the fitted plane and the reference plane

CONCLUSION

In this paper, we have performed a complete virtual photogrammetry measurement of an object by using an optical simulation software package (FRED[®]) and a photogrammetry software package (PhotoModeler[®]). The error of the virtual photogrammetry measurement is at a level of less than 1 part in 10^3 . Preliminary investigations show that errors introduced by the optical simulation process itself are approximately 10 times smaller. These results suggest a potential application of optical simulation packages in uncertainty and sensitivity assessment of photogrammetry measurement in the future.

ACKNOWLEDGEMENTS

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HIGH POWERED FREQUENCY STABILIZED HENE GAS LASER WITH 3.5 MW FROM A SINGLE MODE

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INTRODUCTION

Helium Neon (HeNe) gas lasers are widely used in research and industrial applications because they are directly traceable to the meter [1, 2], have a narrow linewidth, have high coherence, and can be frequency stabilized via relatively simple techniques. Stabilized HeNe lasers are frequently employed as the source for displacement interferometry applications, most requiring low optical power (around 1 mW). However, as the optical system complexity increases and more fiber-based, remote delivery systems are developed, fiber insertion losses can significantly reduce the total available power to the system. Instead, diode lasers or solid state lasers are used which have significantly higher output power but either suffer from a high capital cost for the laser or have insufficient linewidth, coherence, or frequency stability for the intended application. Conversely, HeNe lasers only need a higher output power out of a single mode to further their practicality in future optical systems. The output power of stabilized HeNe lasers has largely been limited because the frequency stabilization techniques require lasers with at most two modes present.

In previous research, we demonstrated a frequency stabilization technique which used a HeNe laser with three modes nominally symmetric within the gain profile [3]. By using a longer laser tube and a large cavity, the three-mode laser has a higher output power and the center of the gain curve with the highest power is used. From Lamb [4], the three mode HeNe laser has three main modes, along with three additional modes with a lower power, each near a main laser mode. A schematic of this is shown in Figure 1. The longer laser tube provides a higher overall output power and specifically a higher output power by the central mode. The alternating polarization states between two consecutive main

modes means the main central mode and smaller central mode can be isolated via polarization. The slight frequency difference (below 1 MHz) between the main central mode and the smaller central mode can be detected and used for feedback stabilization.

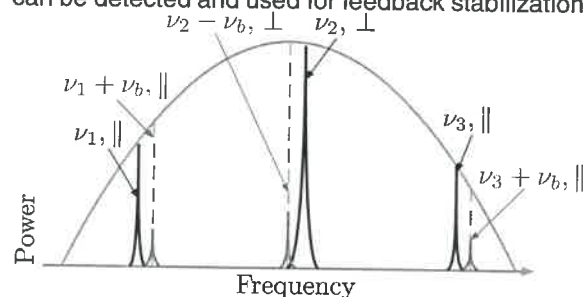


FIGURE 1. Schematic of the three main modes and three mixed modes when nominally symmetric within the HeNe gain profile. Consecutive laser modes are orthogonally polarized and the smaller modes have the same polarization state as their nearest main mode. Because of this, ν_2 and $(\nu_2 - \nu_b)$ can be isolated with a polarizer or optical isolator.

THREE MODE STABILIZATION

Lamb [4] first described the model of the three mode laser where, when nearly symmetric in the HeNe gain profile, the three mode laser contains three additional low power modes. The three additional smaller modes arise from third-order nonlinear harmonics and their exact frequency difference with respect to their nearest adjacent mode depend on cavity tuning and excitation state of the active medium. In this work, the frequency difference between a smaller mode and the main mode, ν_b , ranged between 200 kHz and 550 kHz. As shown in Figure 1, the three main modes from lowest frequency to highest are ν_1 , ν_2 , and ν_3 and the three smaller mode frequencies are $(2\nu_2 - \nu_3)$, $(\nu_1 + \nu_3 - \nu_2)$, and $(2\nu_2 - \nu_1)$. Alternative expressions for the three smaller modes are $(\nu_1 + \nu_b)$, $(\nu_2 - \nu_b)$, and $(\nu_3 + \nu_b)$,

respectively.

The locking signal is obtained by isolating ν_2 and $(\nu_2 - \nu_b)$ using a polarizer or optical isolator because successive modes of a laser without Brewster windows have orthogonal polarization states. The output beam is sampled and ν_b is detected with a low speed photodiode (<600 kHz) and used for feedback stabilization. As the laser frequency changes due to thermal fluctuations in the cavity, the frequency of ν_b changes because the locations of the three modes (and subsequent mixed modes) change. The laser frequency is stabilized by detecting the frequency changes of ν_b relative to a microcontroller clock and stabilizing those changes by thermal actuation chosen here for practical reasons with the given HeNe laser. The feedback system consists of a custom analog interface board, a microcontroller prototyping board employing a PID controller, and a custom current amplifier driving a peltier device which changes the laser cavity length by heating or cooling the laser tube.

Two different lasers were measured during these experiments. Laser A is a nominally 633 nm HeNe gas laser with a 687 MHz FSR (25-LHR-121, Melles Griot). The nominal output power of Laser A is around 2 mW and operates with two to three longitudinal modes within the lasing gain medium. Laser B is a nominally 633 nm HeNe Gas laser with a FSR around 430 MHz (3225H-PC, Hughes). The nominal output power of Laser B is greater than 6 mW and operates with three to four longitudinal modes within the lasing gain medium. Laser B is almost long enough to support five longitudinal modes when the central mode is nominally aligned to the center of the gain medium.

LASER A MEASUREMENT RESULTS

During laser warm-up, numerous mode hops occur as the glass laser tube expands from thermal expansion. Successive modes have orthogonal polarization states and the polarizer/isolator only blocks one polarization state. Thus, the mixed mode signal detected is either from the central mode or from the two outer modes. Figure 2 shows the comparison between successive mixed mode signals where the first mixed mode signal detected is relative to the outer modes and the second mixed mode signal is relative to the central mode. From previous research [3], the laser can be stabilized in both regions but the preferred configuration is with the central mode. The absolute frequency shift is measured relative to a standard iodine stabilized laser. The outer mode configuration has two different optical

frequencies mixed with the iodine laser. Interfering with two modes causes triggering errors which is why the measured absolute frequency shift is almost 1 GHz when the total absolute frequency shift should be less than the FSR (687 MHz).

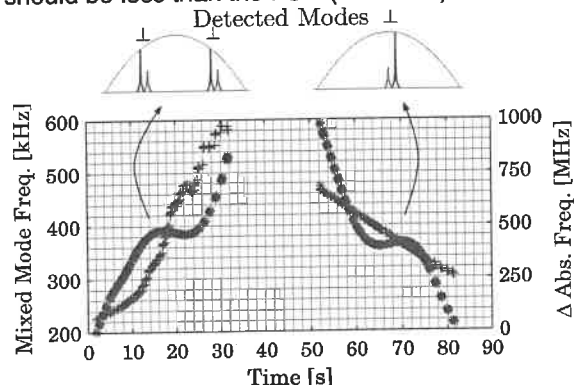


FIGURE 2. Mixed mode signal and absolute frequency shift for successive mode hops during laser warm-up. The left pair is from the outer due to the polarizer whereas the right pair is the central mode.

Laser A nominally outputs around 2 mW, including all modes. In our specific implementation, the central modes are isolated with an optical isolator which reduces the potential power slightly but also prevents optical feedback destabilization. After the unwanted polarization state is blocked and a beam sampler splits a part for feedback detection, 1.78 mW of output power was available for use.

The optical power of the central mixed mode was measured using a DC level detector while the laser was stabilized to minimized drift effects. The amplitude of the AC component was 9 mV and the DC component was 1.8 V. Thus, the optical power of the central mixed mode was approximately 0.5%, or 8.9 μ W. While small, this value should be considered specifically for heterodyne interferometry applications because this signal can contribute to periodic nonlinearity. One method to mitigate this is to use a heterodyne frequency far below or far above the nominal locking signal.

The laser frequency was stabilized using the mixed mode signal for feedback relative to a microcontroller clock. The microcontroller generated a PID signal which was sent to a current amplifier to drive a peltier device. The change in temperature across the peltier caused the cavity length to change via thermal expansion which changed the absolute frequency. Thus, by stabilizing the feedback signal, the absolute laser frequency was stabilized. Figure 3 shows the bandwidth limitations of the thermal con-

troller where the fluctuations were too fast to control. However, by measuring and subtracting this correlated effect, it is possible to improve the absolute frequency stability measurement. In this figure, the laser was stabilized to the mixed mode signal at 300 kHz. The red line in the figure shows the absolute frequency shift from the between the three-mode laser and the iodine laser. The blue line is the measured mixed mode signal multiplied by the absolute frequency sensitivity of 684 Hz/Hz (from Figure 2). Both signals are highly correlated and the difference is shown in black, offset by 100 kHz.

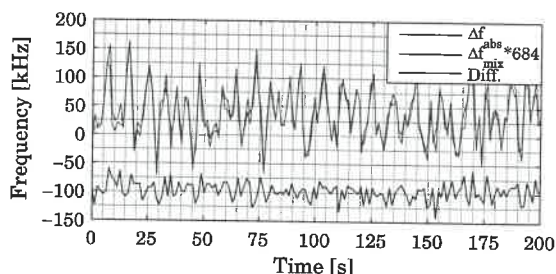
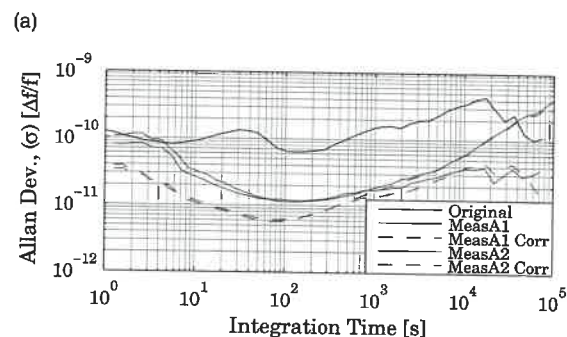
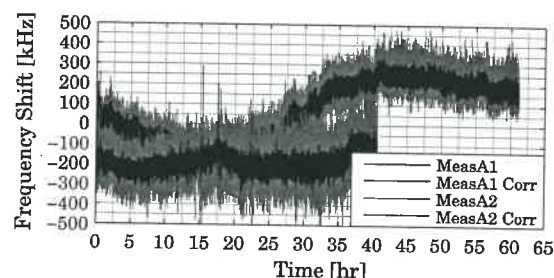


FIGURE 3. (color online) Comparison between the absolute frequency shift (red) and the mixed mode signal (blue) multiplied by the 684 Hz/Hz sensitivity coefficient. The high correlation indicates this signal is useful as a feedback signal. The signal fluctuations are faster than the closed loop controller bandwidth. The difference between the two, essentially the error in the correlation coefficient, is shown in black (offset by 100 kHz for clarity).

Two longer measurements were made with the updated controller relative to the iodine laser. The results from those measurements are shown in Figure 4a. The wide band is the directly measured absolute frequency shift. The narrow band within the measurement is the absolute frequency shift corrected for known signal perturbations. The frequency shift of corrected first measurement was 296 kHz, 2σ (6.2×10^{-10}) with a correlation coefficient of 0.849 between the mixed mode signal and the absolute frequency fluctuations. The second measurement was better; 75 kHz, 2σ (1.6×10^{-10}) with a correlation coefficient of 0.912 between the two signals. Figure 4b shows the Allan deviation for the initial measurement, as well as the uncorrected and corrected measurements after updating the controller. The short term stability for the three uncorrected measurements are nominally the same, around 1 part in 10^{10} . Over hourly and daily cycles, however, the updated controller significantly improves the laser's stability by an order of magnitude. Compensating for the known signal pertur-

bations can also improve the short term stability by roughly a factor of two, to a minimum of 6×10^{-12} (2.8 kHz) at an integration time of 71 s.



(b)

FIGURE 4. (a) Two longer measurements of the absolute frequency shift after updating the controller. The darker bands inside the wider bands is the frequency fluctuations after correcting for known signal perturbations oscillations. (b) Allan deviation of the original measurement (black) and the two measurements with the updated controller (red, blue). The corrected measurements (dashed) have a significantly lower short term Allan deviation.

LASER B MEASUREMENT RESULTS

Laser B, which has a much longer tube, was tested to observe whether the mixed mode signal has the same characteristics and profile relative to the iodine laser as the Laser A. Laser B has a FSR around 430 MHz and is almost long enough to support five longitudinal modes. Laser B, like Laser A, has no Brewster windows which means it also has alternating polarization states. Because Laser A is much smaller, it nominally has two longitudinal modes and only has three modes when the three mode are nearly symmetrical under the gain profile. Laser B, because it is much larger, has mostly four longitudinal modes except when symmetric, which is when it produces only three modes.

The comparison between the mixed mode signal and the iodine laser (locked to the 'i' hyperfine absorption peak) while Laser B was warming up is shown in Figure 5. The first and last segments show

a linearly increasing mixed mode signal with a linearly decreasing absolute frequency shift. This is the instance where the output mode is the central mode. Once the mode hop occurs, at around 120 s, the laser operates with four modes. The detected mixed mode signal is likely due to a misalignment of the optical isolator. After the next mode hop (240 s), the mixed mode signal decreases linearly but the absolute frequency shift is not consistent. This is because the two outer modes each interfere with the iodine laser, causing triggering problems in the frequency counter.

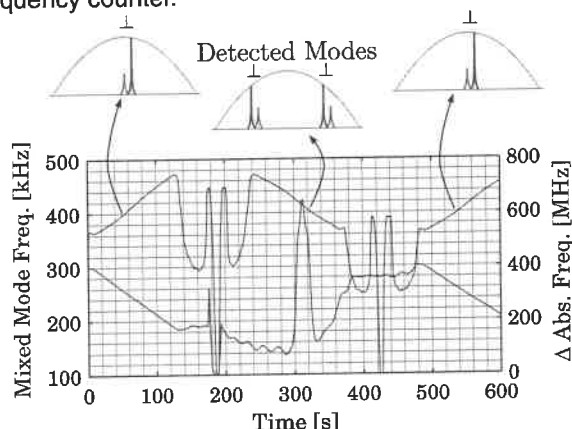


FIGURE 5. The measured mixed mode signal and absolute frequency shift for successive mode hops during warm up for Laser B. The first and last segments have linear locking regions with nominally linear absolute frequency shifts.

The absolute frequency of Laser B was stabilized using the same controller and peltier as Laser A. Because the controller was not designed for this system, the performance was about an order of magnitude worse than with Laser A. By multiplying the measured mixed mode signal by the sensitivity of -1641 Hz (absolute) per Hz (relative), the two signals are highly matched, as shown in Figure 6a. Removing the known fluctuations which could be mitigated with a faster controller, the absolute frequency shift was better than 5 MHz pk-pk over 110 minutes. The Allan deviation for the laser frequency stability is shown in Figure 6b. The frequency stability is approximately one order of magnitude worse than for Laser A, even when correcting for known fluctuations. This difference is likely caused by the longer laser tube and the environmental fluctuations acting over a larger area.

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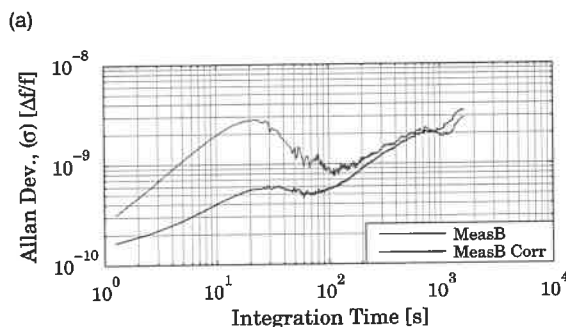
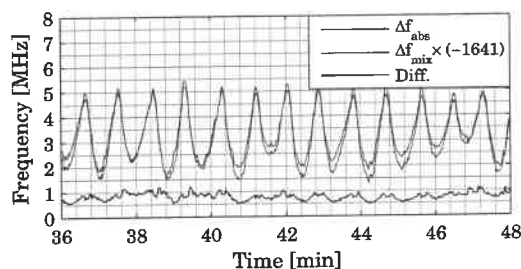


FIGURE 6. (a) Close-up comparison of the measured absolute frequency shift and correlated mixed mode fluctuations after multiplying by -1641 Hz per Hz. The 4 MHz fluctuations were reduced to approximately 1 MHz, which means significant improvements to the controller can improve the frequency stability. (b) Allan deviation of the measured absolute frequency shift (raw) and after correcting for known fluctuations (corr). The short term fluctuations are limited by the speed of the controller and thermal time constant while the long term fluctuations are larger driven by environmental effects due to the laser's size.

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ASPHERIC WAVEFRONT MEASUREMENT BY REFERENCE-FREE WAVEFRONT SENSOR (RFWS)

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INTRODUCTION

Significance and Difficulties of Aspheric Wavefront Measurement

Aspheres are becoming more and more widely used for high-end optical systems in various applications because they offer a high number of parameters for simplifying systems while optimizing their performance. This can lead to systems with fewer surfaces, less weight, smaller dimensions, and high states of correction [1].

Advanced manufacturing technologies allow a wide variety of shapes to be created with optical surface quality. However, such advances are partly limited by poor metrology infrastructure. Till now, the measurement and characterization of such surfaces is still very difficult and open to interpretation [2].

Shack-Hartmann Wavefront Sensor

The traditional Shack Hartmann wavefront sensor (SHWS) measures a wavefront by comparing the images of the focal point matrix of the sample and a reference piece [3]. The centroid displacement reveals the slope of the wavefront to be measured at corresponding sampling position. Wavefront reconstruction can then be applied to get the wavefront based on the slope matrix calculated.

Compared with interferometry-based wavefront sensing technique, the SHWS has a much larger dynamic range [4], although it has some fundamental limitations in measuring highly aberrated wavefronts [5]. One of the main limiting factors is the need of a reference piece [6] due to the many problems it introduces. Firstly, in many scenarios, it is impossible to find such a golden piece that is of similar form [3] and material properties [1] with that of the sample. Secondly, the repositioning of the two pieces exactly at the same spatial position is still difficult to achieve. Last but not least, the

maintenance and operation of the system with an updated and valid reference is troublesome. To solve these problems, we have proposed a new methodology in which wavefronts are measured without a reference, and therefore, the SHWS technology is extended into a reference-free wavefront sensor (RFWS).

APPLICATION OF RFWS IN ASPHERIC WAVEFRONT MEASUREMENT

Methodology

The basic idea of RFWS is to introduce a small lateral disturbance to the sample, and the centroid displacement of each sampling aperture reflects the change of the wavefront slope at that local sampling position. Thus, the wavefront slope changes along x and y direction at each aperture position are obtained by introducing lateral disturbances along x and y direction respectively. Then the wavefront slope matrix is reconstructed by applying a suitable iterative method. At this moment, we adopt Southwell algorithm for reconstruction [7]. The wavefront is subsequently reconstructed with the wavefront slope matrix.

Current wavefront sensing technology measures wavefronts only with small distortions about a flat plane, where a flat reference is readily available, and therefore not suitable for measurements of aspheric surfaces and aspheric wavefronts since the desired reference is difficult to be achieved. On the contrary, RFWS measures a wavefront without the need of a reference. This provides the possibility of complicated form measurements, such as aspheric and freeform, practically possible, so long as the system can sample the surface effectively.

Simulation

By using the virtual RFWS platform established in the Matlab environment, several typical

aspheric wavefronts are measured. Fig. 1 compares the nominal wavefront and the wavefront obtained after two-step reconstruction. The well agreement of the shapes between the reconstructed wavefront and the initially designed formula shows the capability of applying the RFWS in aspheric wavefront measurement.

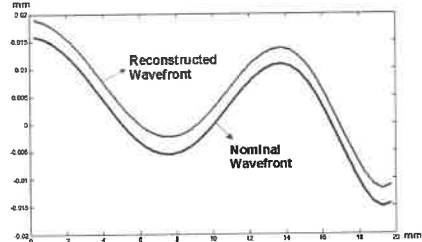


FIGURE 1. Comparison of the nominal wavefront and the curve obtained in the simulation platform.

Experiment

We also measured an aspheric surface with a shape similar to Fig. 1(b) in our RFWS system. The result plotted against the traditional SHWS measurements are shown in Fig. 2. From the fitted coefficients summarized in table 1, we conclude that the form achieved with RFWS is comparable with that from the traditional SHWS.

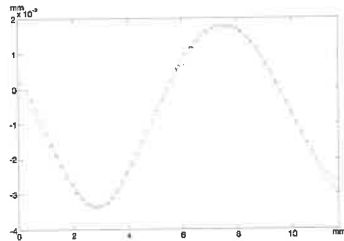


FIGURE 2. Experimental data points obtained after two-step reconstruction. Red stars: measurement data from RFWS system. Blue circles: measurement data from the traditional SHWS system.

TABLE 1. Form reconstructed from the two measurement systems and comparisons of each fitted coefficient (SH: SHWS; RF: RFWS). Error is calculated by the percentage difference with respect to the measurement result from SHWS.

	X^5 (10^{-8})	X^5 (10^{-6})	X^4 (10^{-5})	X^3 (10^{-4})	X^2 (10^{-3})
SH	-5.54	3.63	-8.39	8.53	-3.74
RF	-6.08	3.89	-8.79	8.74	-3.71
Error (%)	9.83	6.98	4.74	2.52	0.76

CONCLUSION

In conclusion, we have demonstrated the effectiveness and feasibility of the proposed RFWS system in measuring aspheric wavefronts by both theoretical simulation and experiment. The reference free wavefront methodology is promising to broaden the applications of S-H wavefront sensors in measuring wavefronts where no golden pieces are fabricated yet, and hence provide the potential to measure aspheric wavefronts. In the future, we will work out a calibration method and study affecting factors in the RFWS system in great detail.

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SHACK-HARTMANN WAVEFRONT SENSOR WITH DIGITAL SCANNING

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LIMITED LATERAL RESOLUTION OF THE TRADITIONAL SHACK-HARTMANN WAVEFRONT SENSOR

In the traditional Shack-Hartmann wavefront sensing (SHWS) system, each lenslet acts as an optical probe. An array of probes samples the surface to be measured and each probe carries the information of the surface's corresponding local area. This information is reflected in the centroid position at the detector plane. Hence, in order to generate more sampling points so as to increase the lateral resolution, more probes are needed to sample a surface. However, due to the following reasons, there is a limitation to the maximum number of probes that a system can provide.

Firstly, an increase in the number of probes indicates a decrease in the diameter of each lenslet. However, the size of the lenslet is limited within a certain range due to both engineering capability and diffraction effect, which makes correlation of each spot at the detector plane with its corresponding lenslet ambiguous [1].

Secondly, the accuracy of SHWS is affected by the focal spot centroid finding accuracy [2]. Generally speaking, for a fixed focal length, bigger the lenslet size, smaller the focal spots, and thus higher the accuracy [3]. As such, it is desired to increase the diameter of the lenslet, so as to reduce the spot size [4].

Last but not least, when SHWS is applied to measure highly aberrated wavefront, such as aspheric surfaces, which is of high demand today, the number of lenslets is further limited so as to provide a large dynamic range of measurement [5]. In other words, crosstalk is more likely to happen when there are more focal spots [6]. Therefore, there is a contradiction between the high lateral resolution achieved by increasing the number of lenslets and measurement range.

To summarize, from physical configuration, measurement accuracy, and application points of view, improving the lateral resolution by purely increasing the number of lenslets is difficult to be realized.

SHWS WITH DIGITAL SCANNING

Methodology

In review of the literatures, we propose a digital scanning SHWS. Without using a physical lenslet array, a spatial light modulator (SLM) is used as the sampling aperture [7, 8]. The lenslet array pattern generated by SLM is laterally shifted digitally to achieve scanning (Fig. 1). The images obtained are stacked together. Wavefront reconstruction is then applied on this resultant compiled image.

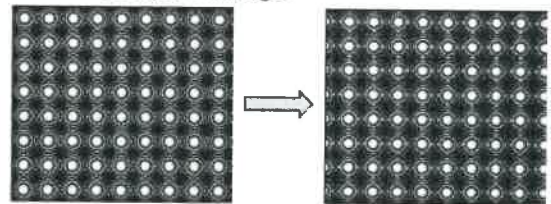


FIGURE 1. Digital scanning of the lenslet array pattern realized by a spatial light modulator.

The improvements of the proposed method over the traditional SHWS include the followings:

- a) The number of sub apertures in each image and the total number of sampling points can be easily adjusted.
- b) Hundreds or even thousands sampling points are possible, and hence the lateral resolution is greatly increased.
- c) The increase in the sensitivity does not sacrifice the dynamic range, which remains the same as that determined by the lenslet array pattern generated in each image frame.
- d) Since the whole pattern is laterally shifted, all sub apertures are active in each image,

which makes full use of the sampling aperture.

- e) Due to the programming capability of SLM, each probe can be individually designed according to the specific requirements of different applications.
- f) No mechanical moving parts are present in the system.

Experiment

Experiment has been done and the system's capability in increasing the lateral resolution is demonstrated where the SLM is designed to generate a lenslet array pattern with diameter of 1.6mm, and focal length 90mm and the CCD receives 9 focusing spots in one image. The result reconstructed from the 9-spot data does not follow well with the surface's true profile as shown in Fig. 2. After conducting 5-time scanning, which generates 45 focusing spots in total, the newly reconstructed result agrees well with the true surface profile.

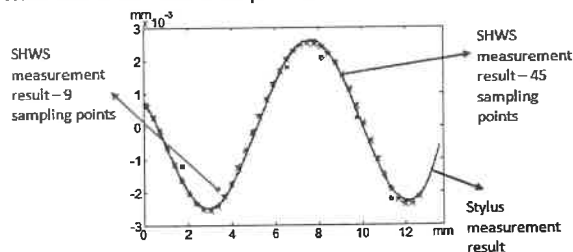


FIGURE 2. Importance of scanning proved in experiment.

NUMERICAL INVESTIGATION OF SYSTEM PARAMETERS

In the process of conducting scanning, several parameters that affect the system's performance are observed. To study these parameters, a simulation platform of the SHWS system (Fig. 3) is developed in the Matlab environment. A collimated testing beam is projected onto the surface along the optical axis, and then reflected to the lenslet array as the sampling aperture, which is formed by plano-convex lenses. The beam is then sampled and refracted when passing through the two faces of each simulated lenslet, which subsequently reach onto the CCD plane. The light propagation of each ray in the system till intersection with the CCD plane is calculated and simulated by optical ray tracing. In this simulation platform, the sample surface to be measured, which is simplified as a 2D curve, is fixed at 16mm. The surfaces simulated are in the shape of sine waves. The peak-valley (PV) value is fixed at 10 μ m. In consideration of

possible occurrence of crosstalk, the scenarios for 0.25 period (0.25H), 0.5 period (0.5H), 1 period (1 H), 1.5 periods (1.5H), and 2 periods (2H) are chosen to be examined.

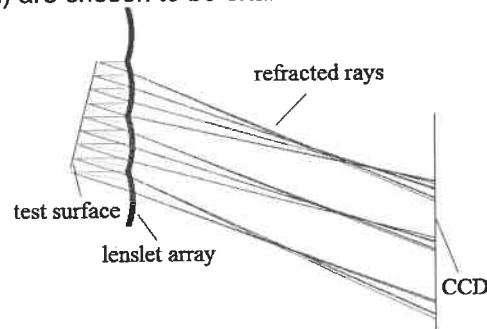


FIGURE 3. Simulation platform.

Focal Length

It is observed that the focal length has little influence on the system performance. For example, in the case of using lenslet array with diameter of 50 pixels to sample the surface with 2 periods, when the focal length changes from 100mm to 1000mm, the corresponding root mean square (RMS) error, ϵ_{rms} , changes from 0.726 μ m to 0.727 μ m. The difference is as small as 1nm. Hence, the contribution of varying focal length on the overall accuracy is negligible. As such, the focal length is fixed at 100mm when study other system parameters.

Lenslet Diameter

In the simulation platform, lenslet diameters of 10 pixels, 15 pixels, 25 pixels, 30 pixels, 45 pixels, 50 pixels, 75 pixels, 90 pixels, and 150 pixels have been studied. In order to make fair comparison on how the lenslet size affect the system's performance, the minimum ϵ_{rms} is defined as the error thus resulted when the total number of sampling points after scanning is 900, which is the maximum allowable number of points that can be achieved from the lenslet sizes that have been studied by the current experiment setup. The corresponding minimum ϵ_{rms} of the reconstructed wavefront generated at various lenslet sizes is recorded and compiled in Fig. 4.

From Fig. 4, it is observed that regardless of the number of periods the surface has, the minimum ϵ_{rms} increases as the size of the lenslet increases. This can be explained from the aspect of the physical model of SHWS. Each focusing spot at CCD can be thought as the averaging information from a surface area that is determined by the size of the lenslet. Big lenslet size implies that each point represents the

averaging information from a large local sampling area. In other words, when lenslet decreases in size, the surface area represented by each spot decreases, and the ability of each point to describe that part of area increases. That is why for the same number of total sampling points, ϵ_{rms} is smaller for small lenslet diameters, or in short, the sensitivity is higher.

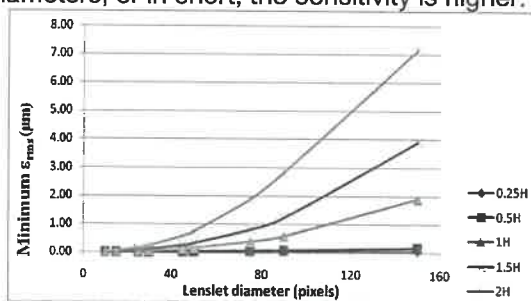


FIGURE 4. Minimum ϵ_{rms} resulted with respect to each lenslet size in various surface form measurement.

Measurement Range

For a fixed lenslet array setting, when the surface to be measured is made up of a higher spatial frequency, the corresponding minimum ϵ_{rms} will be bigger. To illustrate this, the system with a 45-pixel lenslet diameter is taken as an example (Fig. 5). It is seen that when the largest measurable slope of the surface increases from 0.05625 degree (0.25 period) to 0.45 degree (2 periods), the respective ϵ_{rms} increases from 6.1nm to 568nm.

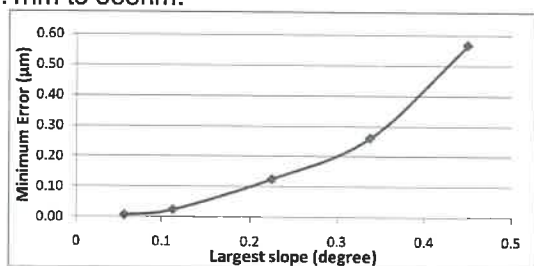


FIGURE 5. Minimum ϵ_{rms} resulted with respect to surfaces with different largest slope, when the lenslet diameter is 45 pixels.

The phenomenon described in Fig. 5 can be explained from the perspective of the physical meaning of the sampling points. Large surface slopes imply that the effect caused by the optical power of the corresponding local sampling area becomes obvious. The effective focal distance deviates from the focal length of the lenslet array, and much greater aberration introduced in the results. As the detector plane is fixed at the focal length of the lenslet array so as to enable the use of the flat plane as the reference, the

smaller the largest measurable slope, or the surface more approaches a flat plane, the higher the accuracy.

Optimum Scanning Step

From mathematical point of view, more sampling points within a fixed surface area mean higher lateral resolution. However, it may not be necessary to have as many sampling points as possible for surface form measurement. Simulation has been conducted to verify the existence of the optimum scanning step, and determine its value in various surface measurements. Here, the optimum scanning step is defined as the value at which the resulted ϵ_{rms} within 10% degradation of the ϵ_{rms} value achieved at the maximum number of sampling points, which is 900 in this simulation environment. In other words, when ϵ_{rms} reduces to 1.1 of the minimum ϵ_{rms} , the corresponding scanning step is considered as optimum. As the CCD used in the experiment has a pixel size of 12μm, and the requirement of efficient generation of lenslet array by the SLM [8], the lenslet diameters of 30 pixels, 50 pixels, 75 pixels, 90 pixels, and 150 pixels have been chosen for study. Take the surface with 1 period as an example, the ϵ_{rms} values thus resulted at various number of sampling points when use different lenslet settings are plotted in Fig. 6.

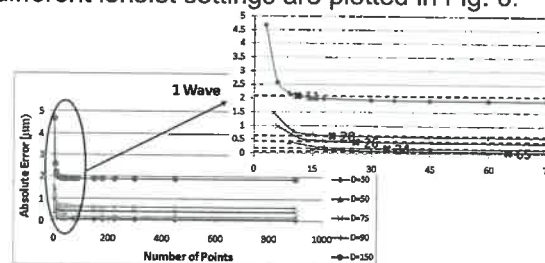


FIGURE 6. ϵ_{rms} for different lenslet sizes and different number of points when the surface has 2 periods.

From Fig. 6, the following relationships are seen clearly:

- For a fixed lenslet size, the error decreases and the accuracy is improved as the number of points increases. However, the improvement in accuracy slows down as the number of points increases, and the accuracy closes to a fixed level for a particular lenslet setting, which is higher for smaller lenslet and lower for bigger lenslet. In other words, in measuring a certain surface, the improvement in accuracy that can be realized through scanning is confined

to a certain level, beyond which it is necessary to reduce the lenslet size.

- b) As discussed above, the optimum scanning step is defined where ε_{rms} reduces to 1.1 of the minimum ε_{rms} . Hence, in Fig. 6, for each series of data, the horizontal value when the vertical value reduces to $1.1 \times \min \varepsilon_{rms}$ is taken down (Table 1) and this is the optimum number of sampling points in that particular scenario. The average value of the optimum scanning step is 4, with a standard deviation of 0.268.

TABLE 1. Summary of the optimum scanning steps for various lenslet sizes.

Lenslet diameter (pixels)	30	50	75	90	150
Optimum number of sampling points	65	34	26	20	11
Number of points per image	15	9	6	5	3
Optimum scanning step	4.3	3.7	4.3	4	3.7

Scanning Strategy

To summarize, it is necessary to choose a suitable lenslet size as the highest level of accuracy a system can reach is intrinsically determined by the lenslet array setting. The conduct of scanning can help to improve the lateral resolution. However, in measuring a certain surface, the improvement in measurement performance that can be realized through scanning is limited to a certain level, beyond which it is necessary to reduce the lenslet size. In the scenarios studied, from efficient and practical points of view, the optimum scanning step is 4.

CONCLUSION

In conclusion, a digital scanning Shack-Hartmann wavefront sensing system to increase the lateral resolution without sacrifice of the sensitivity and dynamic range has been proposed. The digital scanning is done through the implementation of SLM as the lenslet array and the whole sampling aperture is scanned digitally. Through simulation, some important scanning parameters are studied. It is found that for a fixed field of view, the bigger the lenslet

size, the bigger the error of the reconstructed result. In addition, there is a largest measurable slope corresponds to each lenslet size. This measurement range guides us in choosing the reasonable lenslet array setting for measuring different surface forms. Furthermore, the existence of an optimum scanning step eases us in practical application. Although interestingly the optimum scanning steps for the lenslet sizes that have been studied are found to be the same, the level of accuracy is different, due to the intrinsic error brought in by each lenslet array setting. Therefore, it is necessary to choose the suitable lenslet array before proceed to scanning for achieving its practically optimized accuracy level.

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TWO-STEP FOURIER TRANSFORM METHOD FOR INTERFEROGRAM ANALYSIS

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1. INTRODUCTION

The Fourier transform method [1] is an analytical method for interferograms. This Fourier transform method eliminates noise from shape components of the interferograms in the Fourier domain. However, when the noise and shape components overlap each other in the Fourier domain, the accuracy of the Fourier transform method is a problem. Therefore, a method is proposed to solve these problems by using two interferograms with slightly different frequencies.

2. FOURIER TRANSFORM METHOD

The Intensity distribution, $g(x)$ in an interferogram with a tilt is written as follows.

$$g(x) = a(x) + b(x) \cos[2\pi f_0 x + \varphi(x)] \quad (1)$$

where f_0 is the spatial carrier frequency, $\varphi(x)$ represents the desired information of the surface under test, $a(x)$ is the average intensity, and $b(x)$ is the intensity modulation. The intensity distribution described by Eq. (1) may be rewritten as

$$g(x) = a(x) + c(x) \exp(2\pi i f_0 x) + c^*(x) \exp(-2\pi i f_0 x) \quad (2)$$

where

$$c(x) = (1/2)b(x) \exp[i\varphi(x)] \quad (3)$$

and $*$ indicates the complex conjugate. The Fourier transform of $g(x)$ yields:

$$G(f) = A(f) + C(f - f_0) + C^*(f + f_0) \quad (4)$$

where capital letters denote the Fourier spectra and f_0 is the spatial frequency. The Fourier spectra in Eq. (4) are separated by the carrier frequency f_0 . The component of the spectrum centered at f_0 can be recovered without the carrier by the first bandpass filtering and then shifting the isolated spectra back to the origin. This process results in the following equation.

$$G(f) = C(f) \quad (5)$$

However, the filtering process employs the assumption that the Fourier spectra in Eq. (4) are completely separated. The inverse Fourier transform of Eq. (5) then yields the quantity described in Eq. (3). The desired information, $\varphi(x)$ can be determined from the arctangent:

$$\varphi(x) = \tan^{-1} \{ \text{Im}[c(x)] / \text{Re}[c(x)] \} \quad (6)$$

where $\text{Re}[c(x)]$ and $\text{Im}[c(x)]$ are the real and imaginary part of $c(x)$, respectively.

3. PROPOSED FILTERING METHOD

When the component of the desired information and the other components overlap, the desired information cannot be extracted by the bandpass filtering.

The Fourier transform of two interferograms with slightly different carrier frequency f_0 and f_1 produces:

$$G_0(f) = A(f) + C(f - f_0) + C^*(f + f_0) \quad (7)$$

$$G_1(f) = A(f) + C(f - f_1) + C^*(f + f_1) \quad (8)$$

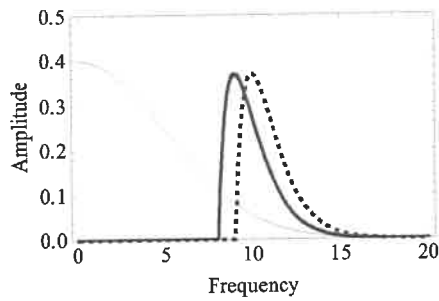


FIGURE 1. Spectral distribution of $A(x)$, $c(f - f_0)$ and $c(f - f_1)$.

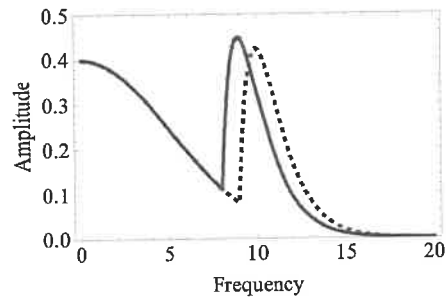


FIGURE 2. Spectral distribution of $G_0(f)$ and $G_1(f)$.

where $G_0(f)$ and $G_1(f)$ are the Fourier spectra of interferograms with carrier frequencies f_0 and f_1 respectively. Figure 1 shows the spectra. The solid curve in red is $c(f - f_0)$, the dotted curve in blue is $c(f - f_1)$, and the thin curve in black is $A(x)$.

In Fig. 2, the solid curve in red is $G_0(f)$ and the dotted curve in blue is $G_1(f)$.

Subtracting Eq. (8) from Eq. (7) yields the following equation:

$$G_0(f) - G_1(f) = c(f - f_0) - c(f - f_1) + c^*(f + f_0) - c^*(f + f_1) \quad (9)$$

Note that unnecessary components $a(x)$ and $b(x)$ have been filtered out of this calculation without experience-based processing. $G_0(f) - G_1(f)$ is shown as Fig. 3. When $f_0 - f_1$ is small, Eq. (9) can be rewritten as:

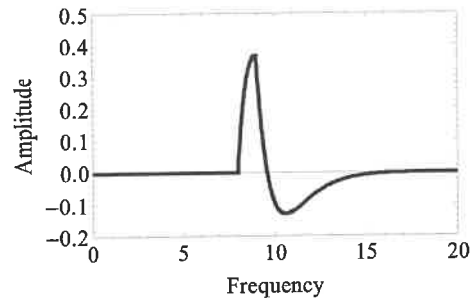


FIGURE 3. Spectral distribution of $G_0(f) - G_1(f)$.

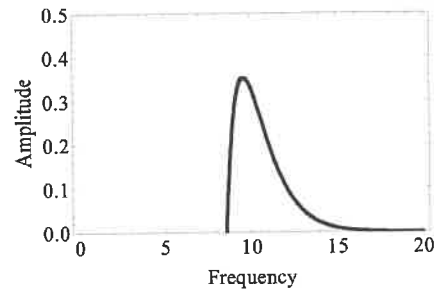


FIGURE 4. Spectral distribution $g(f)$ obtained by integrating $G_0(f) - G_1(f)$.

$$\frac{d}{df} G(f) = \frac{d}{df} c(f - f_0) + \frac{d}{df} c^*(f + f_0) \quad (10)$$

Integrating Eq. (10) with respect to f yields the following equation:

$$G(f) = c(f - f_0) + c^*(f - f_0) \quad (11)$$

In Fig. 4, integrating $G_0(f) - G_1(f)$ in respect to f gives the solid curve in gray. Equation (11) involves only components containing the desired information, $c(f - f_0)$ and $c^*(f - f_0)$.

Elimination of $c(f - f_0)$ or $c^*(f - f_0)$ is straightforward, as $c(f - f_0)$ and $c^*(f - f_0)$ are distributed in the positive and negative frequency domains, respectively.

4. UNDESIRED PHASE INVERSION

The Moiré pattern is formed by superimposing two interferograms with slightly different frequency. Therefore, equation (9) can explain the spectrum of the moiré pattern.

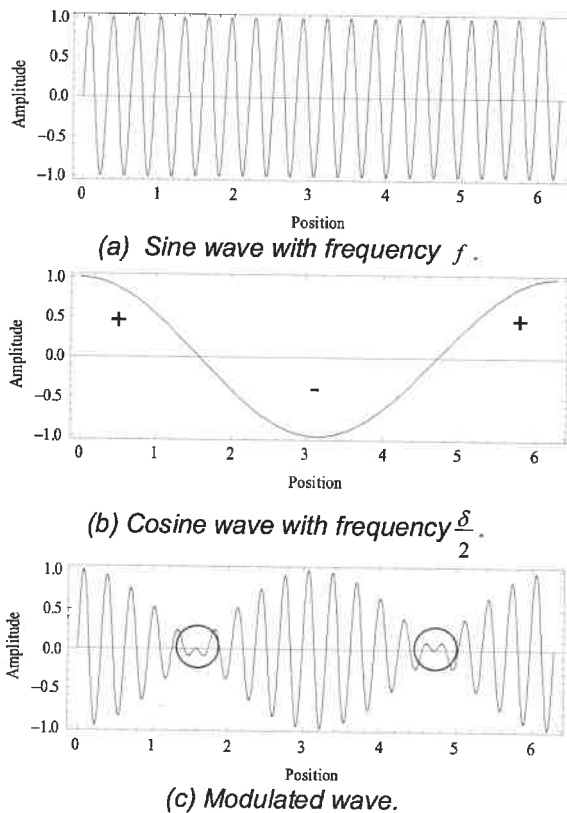


Figure 5 Undesired phase inversion.

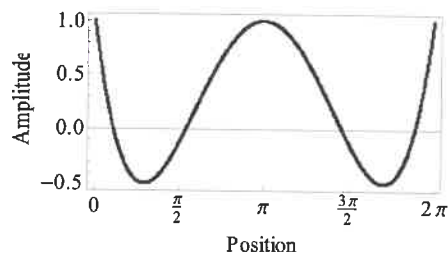


FIGURE 6. The ideal shape.

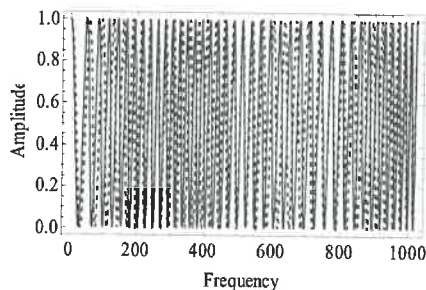


FIGURE 7. Intensity Distribution $g_0(x)$ and $g_1(x)$.

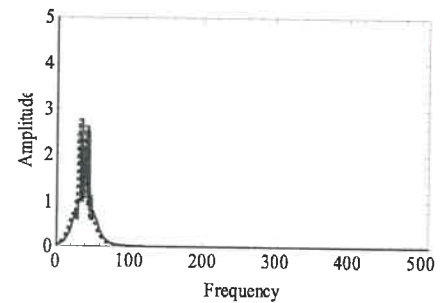


FIGURE 8. Spectrum distribution $G_0(f)$ and $G_1(f)$.

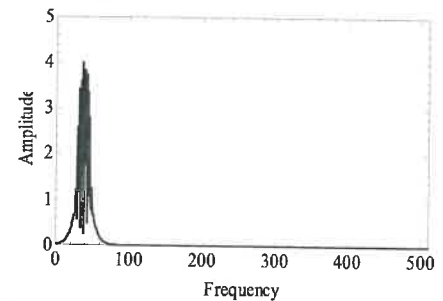


FIGURE 9. Spectrum distribution of $G_0(f) - G_1(f)$.

The intensity distribution $U(x)$ of the moiré pattern is written as follows.

$$U(x) = \sin\left[\left(f - \frac{\delta}{2}\right)x\right] + \sin\left[\left(f + \frac{\delta}{2}\right)x + \theta\right] \quad (12)$$

$$= 2 \sin\left(fx + \frac{\theta}{2}\right) \cos\left(\frac{\delta}{2}x + \frac{\theta}{2}\right)$$

where $f - \frac{\delta}{2}$ and $f + \frac{\delta}{2}$ are spatial carrier frequencies and δ is the difference in the spatial carrier frequency and θ is the phase difference of the sine waves. In short, the moiré pattern means that a sine wave with the frequency f shown in fig.5(a) is modulated by a cosine wave with the frequency $\frac{\delta}{2}$ as shown in fig.5(b).

The intensity distribution $U(x)$ is shown in fig.5(c). Undesired phase inversion, that is, phase shift π occurs in case of $\cos\left(\frac{\delta}{2}x + \frac{\theta}{2}\right) = 0$ at the red circle as shown in fig. 5(c).

4. SIMULATION

In this section, the validity of the proposed filtering method is confirmed by simulation.

In the simulation, the ideal shape is 3rd order spherical aberration as shown in Fig. 6. Figure 7 shows the intensity distributions obtained by the interferometer about the 3rd order spherical aberration with carrier fringe. The solid curve in red is $g_0(x)$ and the dotted curve in blue is $g_1(x)$. The carrier frequency of $g_0(x)$ and $g_1(x)$ are 40 and 36, respectively.

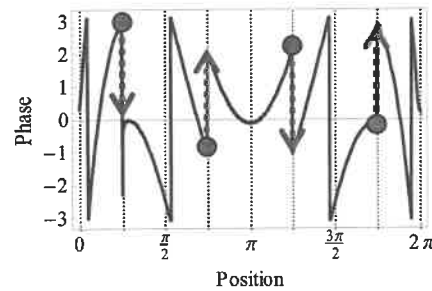
A Fourier transform of $g_0(x)$ and $g_1(x)$ produce $G_0(f)$ and $G_1(f)$. In fig. 8, the solid curve in red is $G_0(f)$ and the dotted curve in blue is $G_1(f)$. Figure 9 shows $G_0(f) - G_1(f)$ obtained by eliminating unnecessary components without the filtering process. Furthermore, to eliminate carrier components, the spectrum was moved toward the origin.

Figure 10 shows phase distributions obtained by the proposed method. There are undesired phase inversions in fig.10(a) and 10(b). In fig.10, the position of the phase inversions are changed in proportion to the phase difference from $g_0(x)$ and $g_1(x)$. Therefore, two analysis results with a phase difference of π are used to avoid the points of phase inversion.

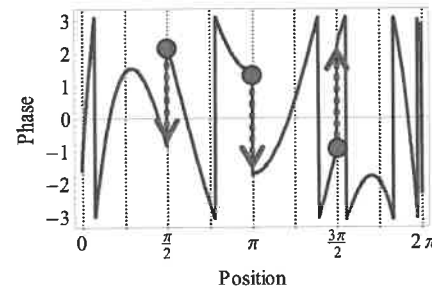
The result obtained by the proposed filtering method as shown in fig.11 is in excellent agreement with the design shape.

5. CONCLUSIONS

In this paper, a filtering method was proposed to eliminate unnecessary components without experience-based processing, and the validity of the proposed method was confirmed by simulation. The result obtained by the proposed filtering method was in excellent agreement with the design spectrum.



(a) A phase difference = 0 .



(b) A phase difference = π .

FIGURE 10. The position of phase inversion.

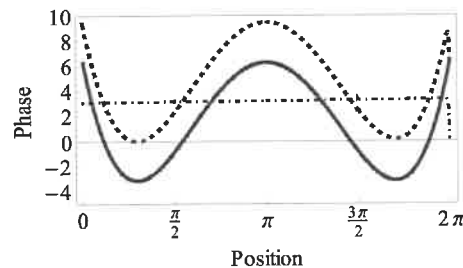


FIGURE 11. A comparison between the ideal shape and the result of phase analysis obtained by the proposed method.

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THE INFLUENCE OF CURING TIME ON HYDROXIDE CATALYSIS BONDING STRENGTH

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INTRODUCTION

Glass-glass and glass-metal bonds are required in many high-precision applications, including displacement measuring interferometers, laser targeting systems, and laser range finders. Common methods include optical contacting and epoxy bonding [1]. Optical contacting, a room-temperature process, is sensitive to surface contamination and can yield unreliable strength. In epoxy bonding, the thickness of interface is relatively large, which can be a concern due to refractive index mismatching between the interface and the substrates. In addition, the mechanical strength of epoxy bonding varies with temperature and chemical environment.

An alternative to these traditional bonding methods is hydroxide catalysis bonding (HCB). HCB was first described by Gwo [2-4] to join the fused silica components which formed the star-tracking space telescope used in the Gravity Probe B space experiment [4]. In this previous work, it was shown that curing time strongly affects the final bonding strength. The purpose of this paper is to determine the time required to reach the maximum mechanical strength using various approaches.

HYDROXIDE CATALYSIS BONDING

HCB is the process by which a hydroxide, such as sodium or potassium hydroxide, catalyzes the surface to be bonded by hydration and dehydration. The HCB technique consists of three steps: 1) hydration and etching; 2) polymerization; and 3) dehydration. The HCB technique enables bonding between a variety of materials if a silicate-like network can be formed at the surfaces. This is the case for most oxide-containing materials, including silica, Zerodur, fused silica, ULE glass, and granite, as well as metals with an oxidized surface.

Several follow-on efforts to Gwo's initial work have further explored the HCB process. For example, Preston *et al.* [5] studied the mechanical strength of BK7-BK7 and silicon

carbide (SiC)-BK7 bonds, while van Veggel *et al.* [6-7] tested SiC-SiC and silicon (Si)-Si bonds. Elliffe *et al.* [8] reported mechanical strength variations based on different types of bonding solution and concentrations of hydroxide ions for various materials. Reid *et al.* [9] explored the influence of temperature and hydroxide concentration on the settling time (i.e., the time required after alkaline solution application before the assembly can be safely removed from the fixture for curing).

EXPERIMENTAL PROCEDURE

The materials used in the experiments were microscope glass slides (water-white, low iron glass) and aluminum samples. The 25 mm × 75 mm × 1 mm thick glass slides were cut into 15 mm × 25 mm sections using a dicing saw. The peak-to-valley (PV) flatness for several samples was measured using a scanning white light interferometer (SWLI); typical values were on the order of 3 μm. A single-side polished 5052 aluminum sheet (300 mm × 300 mm × 1 mm thick) was cut into 15 mm × 25 mm sections using a picosecond micro-machining laser system (Oxford Lasers, Inc., J-355 PS System). Typical PV flatness values for the aluminum samples were approximately 7 μm.

The glass and aluminum pieces were cleaned in an ethyl alcohol ultrasonic bath for 10 minutes and then removed. The remaining alcohol on the surface was removed using an optical wipe and the surface was observed using a magnifier with a high intensity light source. Any remaining particles were removed using a wet ethyl alcohol wipe. Bonding was performed in a class 100 laminate flow clean cube at room temperature to avoid any airborne particles which could degrade the bonding strength. Figure 1 depicts the bonding process using a jig, where the pieces are aligned to have a bonding area of 307.5 mm² (15 mm × 20.5 mm). A sodium silicate bonding solution was used in this study; the solution contained 14% sodium hydroxide (NaOH) and 27% silicon dioxide (SiO₂) dissolved

in de-ionized (DI) water. The bonding solution was dispensed on the top surface of the bottom piece using a micropipette as shown in Fig. 1(A). The second piece was then gently placed on the first piece. See Fig. 1(B). Light pressure was applied to the top piece to uniformly spread the solution. The bonded pieces were left to settle in the jig for about 5 minutes [9] and then moved to another location in the clean cube for curing.

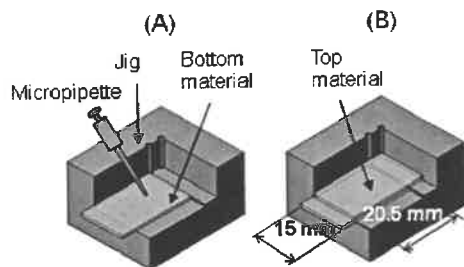


FIGURE 1. The pieces were bonded using a jig for alignment.

Figure 2 shows the test setup for measuring the shear strength of the bonding interface; an axial load frame was used to apply the shear force. A small vise was used to vertically support the bonded sample so that the force axis was parallel to the bonding interface. After a sample was loaded on the lower crosshead, a downward force was applied by the upper crosshead with a speed of 10 mm/min until the sample bond was broken.

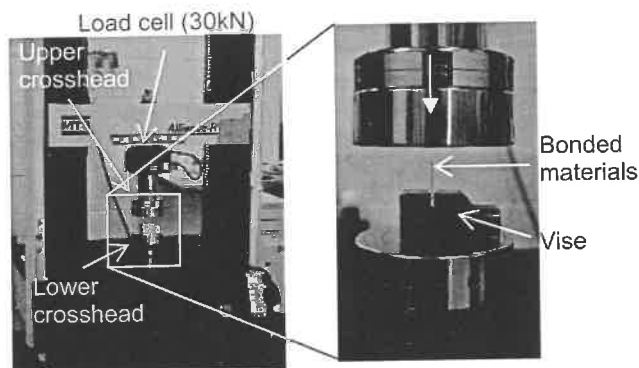


FIGURE 2. Shear strength test setup using an axial load frame with a 30 kN load cell.

RESULTS

As noted, the mechanical strength of the hydroxide catalysis bond depends on curing time. In this section, shear strength results for glass-glass assemblies under various curing times are presented. In an effort to reduce the curing time, the samples were placed in an oven

with elevated temperature to aid the water evaporation. In addition, a commercially-available optical cement was used to bond glass samples for comparison to the HCB results. Since metal components often serve as mounting surfaces and structures for optical bonding applications, the mechanical strength was also tested for aluminum-glass interfaces using the HCB technique.

Glass-glass (HCB)

The influence of curing time on bond strength was tested. A solution amount of 2.0 μ l and 1:4 volume ratio were used. Figure 3 shows the breaking shear strengths for curing times ranging from 1 hr to 5 weeks at room temperature. The bond strength increased with curing time until four weeks. This contradicts the results presented in reference [5] that showed constant shear strength for curing times from 18 hrs to 11 days. However, it supports reference [2], where it was reported that maximum strength was achieved only after 4 weeks of curing time.

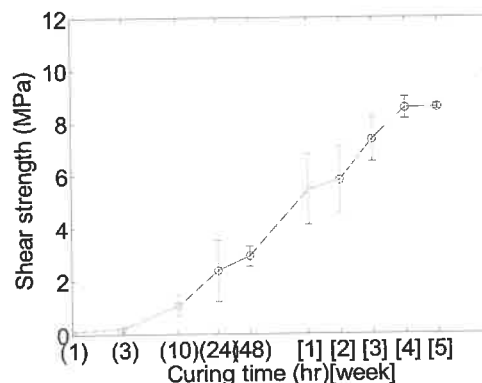


FIGURE 3. Breaking shear strength for various curing times.

The thickness of the bonding interface for a 4 week curing time was measured using the SWLI. The bonded sample was sectioned through its center to reveal the bonding interface and then polished. The interface could not be observed for the 1 μ m lateral resolution of the SWLI measurement. This indicates that the interface thickness was less than 1 μ m.

It was observed that the maximum strength for the glass-glass bonding was obtained after a curing time of 4 weeks at room temperature. However, this time is too long for most commercial applications. Therefore, it was

desired to reduce the curing period. It is known that water migrates or evaporates out of the bond during curing (the final step in the HCB process). In this study, an oven was used to aid in this dehydration step.

Figure 4 shows strength testing results for oven curing at various temperatures. Again, 2.0 μl of 1:4 volume ratio sodium silicate solution was used. The glass pieces were bonded and placed in the clean cube for 24 hrs at room temperature. The bonded samples were then moved into the oven. At each temperature (60 deg C to 100 deg C), the samples were cured from 1 hr to 7 hrs. Ten assemblies were used for each test set. As shown in the figure, the mean shear strengths of the samples cured at 100 deg C for all curing times and at 80 deg C for times of 3 hrs and higher exceeded the maximum strength (solid horizontal line) obtained from the room temperature cure. These tests provided two important results: 1) the curing time for maximum strength can be significantly reduced from 4 weeks at room temperature to 24 hrs at room temperature followed by 1 hr at 100 deg C; and 2) the shear strength of the samples cured at elevated temperatures exceeded the strength of the samples cured only at room temperature.

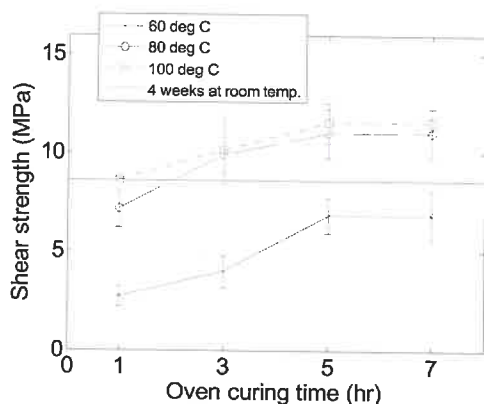


FIGURE 4. Variation in shear strengths for oven curing.

Glass-glass (optical cement)

A commercially-available two-component optical cement (Summers Optical, M-62) was used for glass-glass bond. The manufacturer-specified curing conditions were followed: 1) 30 minutes at room temperature; 2) two hrs at 70 deg C; and 3) 72 hrs at room temperature.

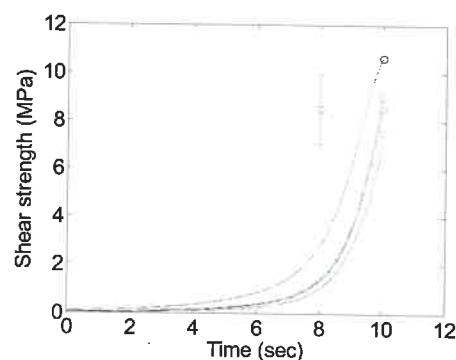


FIGURE 5. Shear strength profiles for glass-glass bonding using a commercially-available optical cement.

The microscope glass slides were again used as materials to be bonded. The bond geometry and breaking process were the same as for the HCB tests. Figure 5 shows the shear strength profiles for the five samples. The mean strength is 8.5 MPa which is close to the maximum strength (8.6 MPa) obtained for the HCB samples cured at room temperature for 4 weeks.

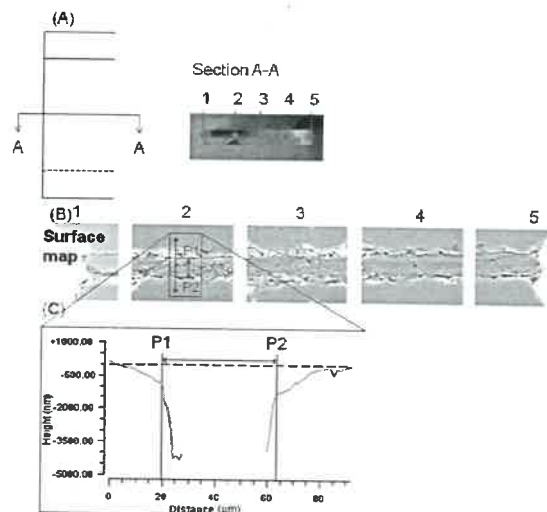


FIGURE 6. Optical cement bond thickness measurements. The mean thickness for the five measurements was approximately 40 μm .

SWLI images of the bond thickness for the optical cement are presented in Fig. 6. The same 1 μm lateral resolution as for the HCB thickness measurement was available. Figure 6(B) shows the bonding interface at each location labeled in Fig. 6(A). A uniform bond thickness is observed. The mean bonding thickness of the five locations was approximately 40 μm as shown in Fig. 6(C). This bond

thickness ($\sim 40\ \mu\text{m}$) is much higher than for HCB ($<1\ \mu\text{m}$), while the bond strengths are comparable.

Aluminum-glass (HCB)

The native oxide layer on metals enables a hydroxide catalysis bond to be formed with a glass sample. Like the glass-glass bond, curing time variation tests were conducted to evaluate the maximum bond strength for aluminum-glass bond. The samples were cured at room temperature for a range of times from 24 hrs to 5 weeks. Sodium silicate solution was again used as the bonding solution with an amount of $2.0\ \mu\text{l}$ ($307.5\ \text{mm}^2$ bonding area) and a volume ratio of 1:4. Five samples were bonded for each curing time.

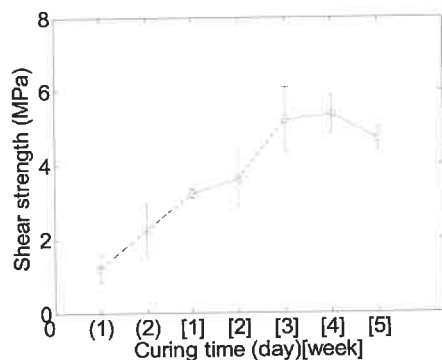


FIGURE 7. Shear strength for HCB aluminum-glass assemblies at various curing times.

Figure 7 shows the shear strength variation with curing time for the aluminum-glass bonds. Maximum strength was obtained after 3 weeks. Although this strength was about half the strength of the glass-glass bonds (when cured at $100\ \text{deg C}$ for 7 hrs), it is reasonable given the large PV flatness value for the aluminum samples.

CONCLUSIONS

In this work, the shear strength for glass-glass (water white, low iron glass microscope slides) and aluminum-glass bonds produced using hydroxide catalysis bonding was evaluated. It was determined that the bond strength increased with extended curing times and elevated curing temperatures. The maximum bonding strength for the glass-glass bond was achieved after 4 weeks at room temperature. The same strength was obtained for a curing time of 24 hrs at room temperature followed by 1 hr at $100\ \text{deg C}$. This shear strength level was

comparable to the results for a two-component optical cement, although the bonding interface of the optical cement was much thicker. Aluminum-glass bonding was also completed. The strength levels were lower due to the relative non-flatness (approximately $7\ \mu\text{m}$ peak-to-valley) of the aluminum samples.

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OPTICAL ALIGNMENT OF SAMM TO REDUCE COSINE ERROR

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INTRODUCTION

The Subatomic Measuring Machine [1-3] has positioning resolution near 10 picometers over a 25 mm X-Y range. In order to reduce the position errors due to the interferometer-laser-mirror misalignments, steps are taken to bring the cosine errors [4] to within a few nanometers across the range of motion. This report gives an overview of the optical techniques used to verify the critical sub-minute laser alignments.

SAMM POSITIONING SYSTEM

The layout of the interferometric measurement system is shown in *Figure 1*. The measurement laser beam enters the system and is split into three beams to provide interference from three interferometers which are used to determine X and Y stage displacements and rotation about the Z axis. There are three capacitance sensors not shown which reference against the top of the mirrored stage to measure rotations about the X and Y axes and also to determine the Z position.

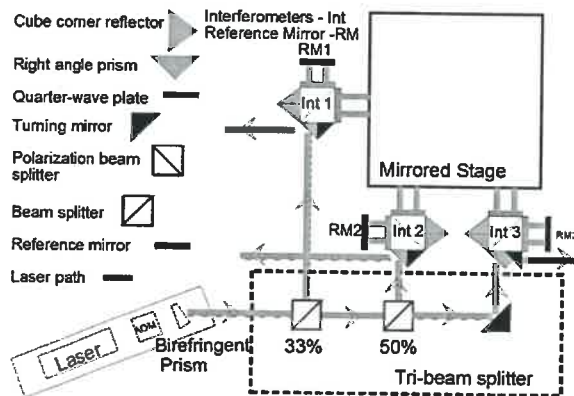


FIGURE 1. Optical schematic of the SAMM laser system.

HOME

Ideally all lasers would be perpendicular to their moveable measurement mirrors when the SAMM is at its 6 degree-of-freedom (DOF) home position. This position is determined by three capacitance gages and three acoustic sensors. However this position can only be

achieved while the stage is under control with the three interferometers operating. Since the reduction of cosine error requires the adjustment of the pose of the interferometers, the problem then becomes one of alignments while the stage is at rest (not under control) but referenced to its home angular pose.

While the stage is at rest the X-Y position and Z-rotation home position can be verified by the acoustic sensors. However the stage is beyond the range of the capacitance sensors in this situation. The capacitance gages had been previously calibrated against a moveable reference flat whose changing pose across the measurement range was verified using an autocollimator. The flat was also initially aligned against the capacitance electrodes of the frame to set the zero angles of the autocollimator. This determines that the surface of the cap gage electrodes should be parallel to the mirrored surface of the stage. We determined the misalignment between these surfaces at rest using the following setup.

STAGE/FRAME X&Y REST ANGLE OPTICS

In order to monitor the X&Y angle relationship of the stage with respect to the metrology frame a laser beam was directed through the Zerodur to reflect from the back of a capacitance electrode on the frame and simultaneously from the top of the mirrored stage. The system schematic and photograph are shown in *Figures 2 and 3*. This setup provided a baseline correction for alignment of the laser to the mirrors while at rest. *Figure 4* shows the separation of the beams projected onto a 3x5" note card at a distance of 7 meters away from the point of incidence. There is a separation of about 10 mm indicating a misalignment of approximately 1.5 milliradians. The spurious reflections are from the top surface of the Zerodur and some edge related diffraction. The two spots are easily distinguished by moving the laser back and forth along the capacitance electrode surface.

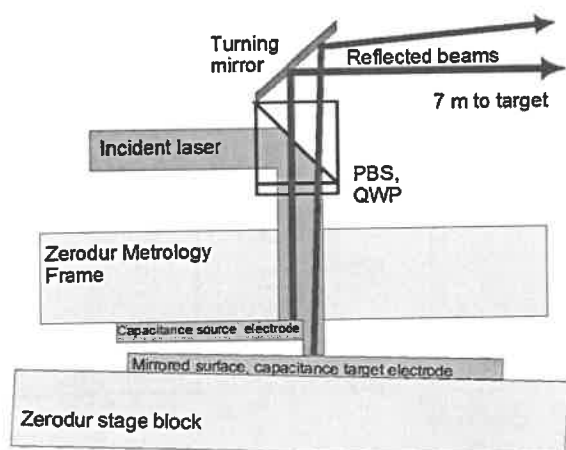


FIGURE 2. Diagram of the optical system for measuring the stage to frame alignment.

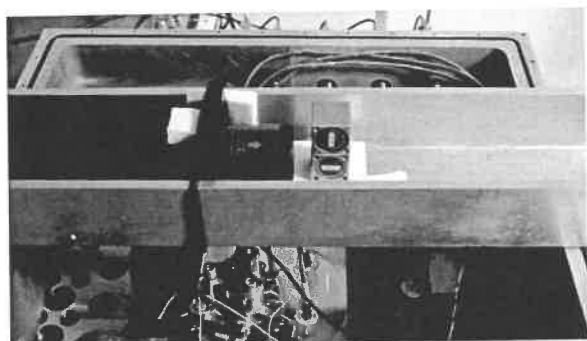


FIGURE 3. Picture of the system measuring the stage to frame alignment.

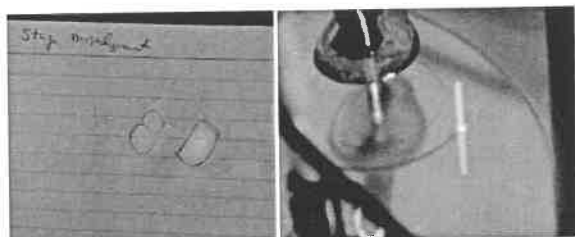


FIGURE 4. Projected laser spots (left) at 7 meters after incidence (right) on capacitance gage electrode and stage top surface.

LASER/STAGE ALIGNMENT OPTICS

Because the laser is situated at a lateral offset from the mirror and passes through various optics, all of which can tilt the beam, it is necessary to both displace the beam laterally (parallel) to the emitted path and also to avoid the optics during initial alignment. This was accomplished through the use of four cube corner retroreflectors shown in Figures 5 and 6.

The laser path is directed to the mirror through an exit port. The laser was rotated using kinematic mounting screws and through two micrometers sliding a sub-plate which supports the laser (not shown).

It is necessary to introduce an optical system consisting of a polarization beam splitter (PBS) and three quarter-wave plates (QWP) at 45° with a beam being retroreflected to compare with the returned measurement beam. The position of the two beams at near and far positions were compared to determine the alignment error. It was also necessary to compensate for the alignment between the stage top and Zerodur frame as mentioned previously. Due to the surface reflections, the cube corners were positioned at an angle with respect to the incident beam in order to clearly separate out the beam reflected from the stage.

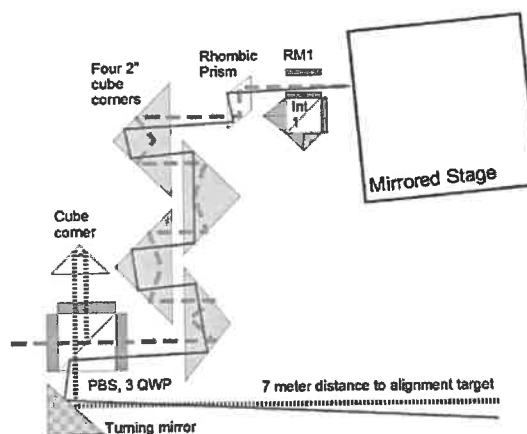


FIGURE 5. Optical arrangement for aligning the laser directly with the mirrored stage surface.

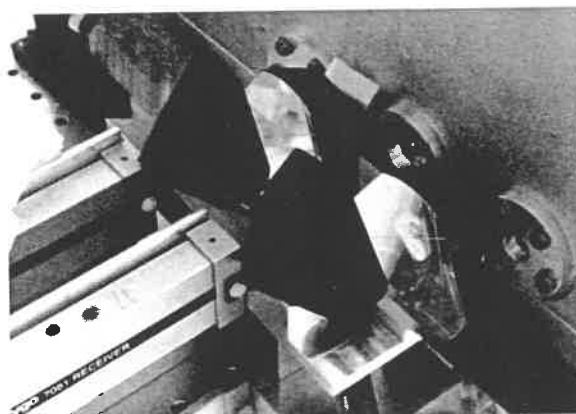


FIGURE 6. Photograph of the four cube corner combination for laterally displacing the laser beam to effect alignment.

TRI-BEAM SPLITTER CO-ALIGNMENT

The reflected beams of the tri-beam splitter shown in *Figure 1* were initially aligned with each other external to the SAMM through the optical arrangement shown in *Figure 7*. The alignments had to be done with the plate in an unclamped state to keep the mounting plate from bending.

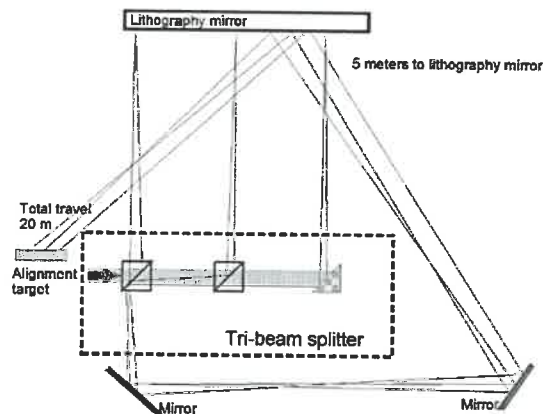


FIGURE 7. Optical arrangement for aligning the splitting optics with respect to each other.

After mounting the tri-beam splitter back into the SAMM, alignment of the splitter to the mirror was accomplished by using a beam reflected from the tri-beam splitter via a route through a rhombic prism to the mirrored stage as shown in *Figure 8* for the 33% splitter. The tilt of the stage mirror at rest also needs to be taken into account to get a proper alignment

The actual setup for determining the alignment of the 33% beam splitter with respect to the appropriate face of the stage mirror is shown in *Figure 9*. It shows that in this case two rhombic prisms were used to get a workable lateral translation. All three splitters can be aligned separately to the stage mirror using an appropriate rhombic prism eliminating the need for the setup in *Figure 7*.

Also shown in *Figure 8* is the setup for aligning the reference mirror RM1 to the aligned laser using two rhombic prisms directing the beam around the Int1 interferometer (after removal of the rhombic prisms used for alignment to the stage).

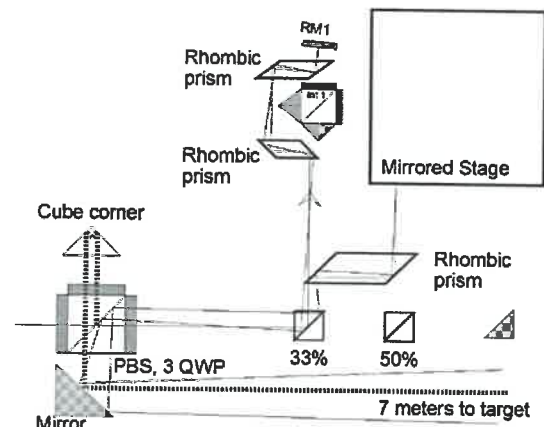


FIGURE 8. Optical setups schematic for aligning the tri-beam splitter to the stage front mirror and also aligning the RM1 mirror to the laser.

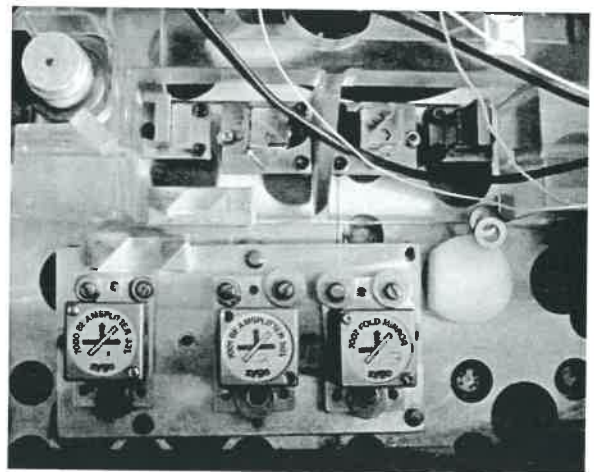


FIGURE 9. Optical setup for aligning the tri-beam splitter (via the 33% splitter) to the stage front mirror using two rhombic prisms.

INTERFEROMETER CONSIDERATIONS

The interferometers used for SAMM reflect the measurement beam from the stage mirror four times before being directed toward the signal detector.

The measurement beam path through interferometers Int2 and Int3 (Fig 1) goes from the tri-beam splitter → PBSt thru → QWP → SM → QWP → PBSturn → CCR → PBSturn → QWP → SM → QWP → PBSt thru → RPM → PBSturn → QWP → SM → QWP → PBSturn → QWP → SM → QWP → PBSt thru → RAM where

QWP – quarter wave plate @ 45°
SM – stage mirror
PBS – a polarization beam splitter
PBSt thru – straight through the PBS
PBSturn – turned by the PBS
RPM – roof prism mirror
RAM – right angle mirror
CCR – cube corner retroreflector

The critical path of alignment is between the interferometer and the stage mirror. The laser needs to be along the path of travel which also needs to be perpendicular to the stage mirror [4].

Tight-tolerances on the orthogonality and flatness of the stage mirror block manufacturing allows the measurement beams to be either orthogonal or parallel if they are aligned to the mirror as well as assuring the orthogonality between the X and Y axes also minimizing the related cosine errors.

If the beams are aligned to the mirror normals (when the interferometer is not in the path) then as the beam makes its first pass through the interferometer and QWP the beam stays aligned and returns aligned. If the PBS then orthogonally reflects the beam in a slightly non-orthogonal direction it doesn't matter as long as the CCR is of sufficient quality to return the beam in the same direction to be corrected in the subsequent reflection and sent through the QWP to the mirror again aligned. Once the beam returns through the PBS to the roof prism mirror (RPM) one direction is returned aligned but the vertical alignment can be compromised and the beam can be sent through the PBS to the mirror not aligned. The return to the PBS is thus also not aligned as it is turned to the CCR and reflected back to the mirror after being

turned by the PBS along the same nonaligned path to be realigned by the mirror. The beam is then directed by the RAM toward the detector. The total result is that two of the beams strike the mirror non-aligned vertically resulting in a cosine error.

Future plans for the alignment include introducing a QWP in the third reflection path to return the beam back along its previous path but having the vertical misalignment to be doubled as it passes through the RPM to be returned nearly along the same line back to a comparison optic as shown in Fig 8 (PBS 3QWP) and then sent to a remote target for comparison. The Int1 interferometer setup will be similar even though the PBS can initially turn the beam to be misaligned to the mirror adding complexity (horizontal misalignment) to the adjustments.

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FREEDOM AND CONSTRAINT ANALYSIS AND OPTIMIZATION

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ABSTRACT

Many mathematical and intuitive methods for constraint analysis of mechanisms have been proposed. In this article we compare three methods. Method one is based on Grüblers equation. Method two uses an intuitive analysis method based on opening kinematic loops and evaluating the constraints at the intersection. Method three uses a flexible multibody modeling approach which facilitates the analysis of complex systems. We demonstrate a visualization method using generalized von Mises stress to show overconstraint modes. A four bar mechanism and a two-degree-of-freedom (DOF) flexure-based mechanism serve as a case study. Briefly the optimization of the location and orientation of releases is discussed. The implementation of the releases in the flexure-based two DOF mechanism is presented.

INTRODUCTION

A system is exactly constrained [1] if the system is kinematically and statically determinate so it has exactly the minimum constraints to kinematically constrain the system. Exact constraint design, as opposed to elastic averaging, does not require tight tolerances on flatness, parallelism and squareness, and it allows for temperature fluctuations without excessive stress in the structure. Elastic averaging (overconstraining) tends to be most successful when machine stiffness is of utmost importance. Overconstrained flexure mechanisms assembled out of several parts are prone to misalignment, which can provoke bifurcation. Meijaard et al. [4] have shown that a misalignment angle of several tenths of milliradians can be sufficient to provoke bifurcation in an overconstrained parallel leaf-spring flexure. The bifurcation results in a stiffness reduction of roughly one order in the intended stiff support directions. It can be concluded that exact constraint design promotes deterministic behavior.

Three methods for analyzing the DOFs and constraints of a system are presented and are demonstrated with the simple example of a four bar-mechanism having one constraint loop. Next a more complicated mechanism like the two-DOF manipulator is presented. It can be considered as several constraint loops. The two-DOF mechanism is designed for xy-motion in vacuum with a relatively large stroke and base mounted actuators.

CONSTRAINT ANALYSIS USING GRÜBLER

In a 2-D analysis using Grübler's [2] equation a four bar mechanism consists of four rigid beams having twelve DOFs in total and four hinges each constraining two DOFs (Table 1). Three DOFs are omitted as they are the three rigid body modes of the four bar mechanism. So in total there is one DOF more than there are constraints. The four bar linkage has one internal mode, the free motion, therefore there are no overconstraints.

Table 1. Constraint analysis of a four-bar mechanism using Grüblers formula.

	2-D	3-D	
4 rigid beams	12	24	DOFs
4 hinges	8	20	Constr.
Rigid mechanism modes	3	6	Constr.
DOFs - constraints	1	-2	DOFs
Underconstraints (motions)	1	1	DOFs
Overconstraints	0	3	Constr.

The mechanism can also be analyzed including the 3rd dimension using Grübler formula's. There are four bars with a total of 24 DOFs, 20 constraints of the 4 hinges, and 6 rigid body modes. In total there are 2 more constraints than DOFs. The 4 bar linkage has one internal mode, the free body motion shown by the 2-D Grübler analysis. Therefore there have to be three overconstraints. The overconstraints do not

appear in the 2-D analysis, but do show up in the 3-D analysis, therefore the constraints have to do with the out-of-plane directions. Three releases are required to obtain an exact constraint mechanism. It can also be concluded that the Grübler equation can be applied correctly only if either the number of underconstraints or the number of overconstraints are known.

CONSTRAINT ANALYSIS BY OPENING A KINEMATIC LOOP

The overconstraints can also be analyzed by opening the four bar loop and examining the constraints as shown in Fig. 1.

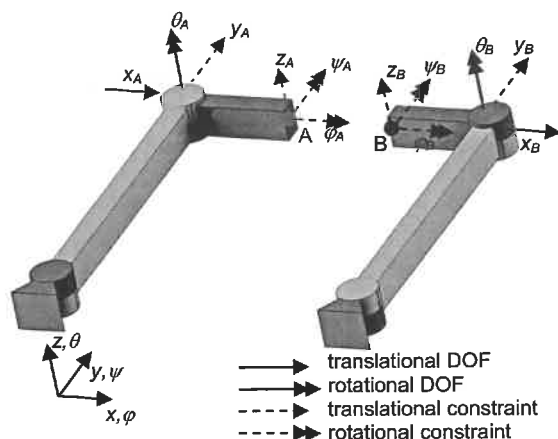


Figure 1. The loop of a four bar linkage is opened. The DOFs (solid vectors) and constraints (dashed vectors) are shown for points A and B.

The in-plane vectors x_A , y_A , θ_A , x_B , y_B and θ_B show two constraints where three are required to be exactly constrained. The linkage has one in-plane DOF. This motion is generally controlled by an actuator system of some kind. In the out-of-plane direction z_A constrains the same direction as z_B . The same holds for φ_A and φ_B , and ψ_A and ψ_B . Three releases should be implemented which lead to the release of z_A or z_B , φ_A or φ_B , and ψ_A or ψ_B to obtain an exactly constrained design.

CONSTRAINT ANALYSIS BY SVD

Using a flexible multibody modeling approach the connection of bodies is described by nodal coordinates; fixed, dependent (calculable) or independent (specified as input variable). In lumped flexure mechanisms each body has a specific number of deformation modes; prescribed (rigid bodies), independent (as input

actuators) or dependent (compliances). In accordance with Grübler [2], the mobility of a mechanism is equal to the number of calculable nodal coordinates minus the number of fixed and independent deformation modes. For a zero mobility the Jacobian matrix, mapping the displacement of the dependent nodal coordinates to the prescribed and the independent deformation modes, must be square. Therefore the Jacobian matrix must also be nonsingular, or equivalently the matrix should have full rank. Only then the mechanism is statically and kinematically determinate. Equivalently the Grübler analysis reveals the difference between DOFs and constraints. The Grübler analysis only gives a correct number of DOFs if the number of overconstraints is known. The matrix rank can be determined from the singular value decomposition. If the Jacobian matrix is row rank-deficient the mechanism is statically indeterminate (overconstrained). The left singular matrix shows a null space which represents an internal stress distribution without external nodal forces being applied [3]. A kinematically indeterminate motion can be derived from the right singular matrix, if the Jacobian matrix is column rank-deficient.

The generalized stress resultants of each element can be converted into an equivalent von Mises stress distribution. The values of the stresses have no physical meaning as they can be scaled. Nevertheless, the distribution shows the locations where stress can be expected due to an overconstraint. Figure 2 shows one of the three overconstraints in loop h_1 - h_4 - h_5 - h_2 , with bending in beams b_1 and b_2 and torsion in beam b_7 . Hinges h_1 and h_2 are fixed to the surroundings.

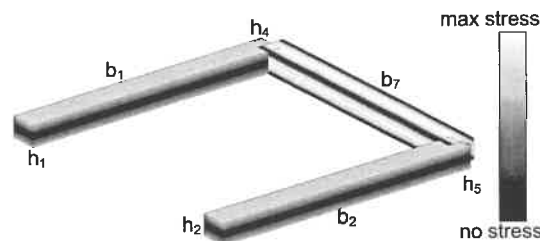


Figure 2. One of the three overconstrained modes of the four-bar mechanism, loop h_1 - h_4 - h_5 - h_2 .

So the SVD analysis not only shows that there are three overconstrained modes in the mechanism, it also shows the accompanying stress distribution resulting from misalignment.

RELEASES OF A FOUR-BAR MECHANISM

There are many configurations possible to release the three overconstraints. One exact constraint configuration is shown in Figure 3 with torsion and twice bending released in beam 7.

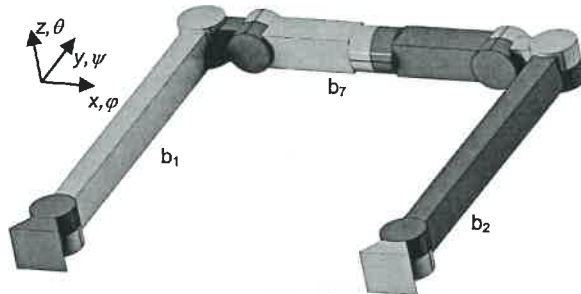


Figure 3. An exact constraint configuration using three hinges in beam 7.

There are three beams with each three locations (bending at both beam ends and torsion) to release the three overconstraints. The number of combinations is equal to the binomial coefficient:

$$\binom{n}{k} = \frac{n!}{k!(n-k)!} = 84$$

with $n = 9$ the number of locations and $k = 3$ the number of releases. By using the SVD analysis it can be proven that of the 84 combinations 44 are exact constraint.

OPTIMIZING THE LOCATION OF RELEASES

Releasing constraints in the case of flexure mechanisms is often done by implementing extra flexures (for small deformations). The location and direction of implementing releases in a mechanism for obtaining an exact constraint design can be optimized for minimization of internal stress and maximization of stiffness (or eigenfrequencies).

A lumped element model has been setup with seven low stiffness hinges, representing the four planar hinges and the three releases. Six stiff hinges are implemented at the remaining possible locations for implementing releases. Each of the 84 combinations has been modeled to calculate the stiffness of beam 7 in z -, φ - and ψ -direction. Misalignment of one of the base connections in z -, φ - and ψ -direction will lead to moments in the hinges causing stress in the beams. The sensitivity of the stress with respect to the misalignment depends on the configuration of the releases. It turns out that only the exact constraint configurations show a combination high stiffness and low stress. This is an expected result. Exact constraint designs are

tolerant for misalignments. The exact constraint configurations showing the highest stiffness allow bending near or in b_7 , not near the base. Several overconstrained configurations show a 27% higher minimum stiffness than the best exact constraint configurations at the cost of several orders higher internal stress.

The direction of releases should be chosen carefully as singular values are dependent on the position of the mechanism. Therefore the mechanism should be analyzed in a variety of positions.

CONSTRAINTS OF THE 2-DOF MECHANISM

The two-DOF xy -mechanism (Figure 4) is designed for base mounting actuators and the use of flexure hinges. The two-DOFs mechanism can be analyzed using Grübler in 2-D and 3-D (Table 2). It should be noted that hinge h_4 and h_5 connect three beams. Therefore they behave like two hinges releasing one DOF for each of the two connected beams. In the case of the 3-D analysis there are seven more constraints than DOFs. Knowing that the mechanism should have two DOFs, the number of overconstraints is nine.

Table 2. Constraint analysis of the two-DOFs mechanism using Grüblers formula.

	2-D	3-D	
9 rigid beams	27	54	DOFs
7 hinges (rel. 1 DOF)	14	35	Constr.
2 hinges (3 conn.beams)	8	20	Constr.
Rigid mechanism modes	3	6	Constr.
DOFs - constraints	2	-7	DOFs
Underconstr. (motions)	2	2	DOFs
Overconstraints	0	9	Constr.

These nine overconstraints are all in the out-of-plane direction. The two DOF mechanism can be interpreted as having three kinematic loops with each loop having three overconstraints like the four bar mechanism. The mechanism can be analyzed by the SVD method. Figure 5 shows an overconstrained mode where six releases have been implemented. Clearly the stress is located in the branch not having releases. Step by step all constraint modes can be resolved by implementing releases.

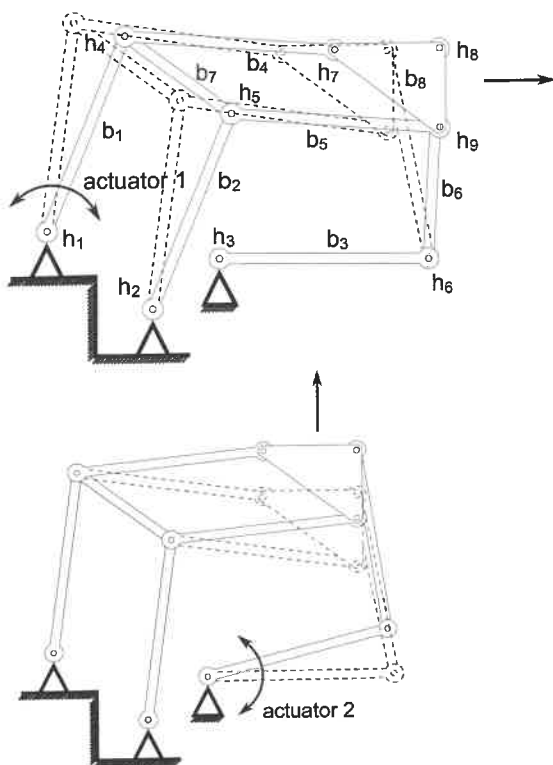


Figure 4. Two DOF mechanism with 2 motions.

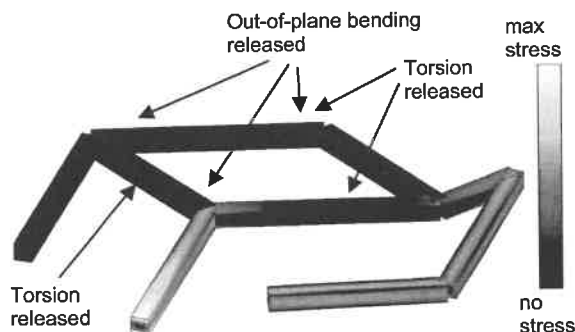


Figure 5. An overconstrained mode of the total mechanism.

OPTIMIZING THE LOCATION OF RELEASES

For the application of this mechanism the first natural frequency with blocked actuators should be as high as possible. Therefore some 9216 exact constraint configurations have been analyzed using the multibody model with SVD analysis and have been ranked on their first natural frequency using a modal analysis. It can be concluded that the best configurations show release of the out-of-plane bending related constraints nearest to the end-effector, the notch pivot flexures in Figure 6. Release of the overconstraint torsion is less critical and is

released in the beams nearest to the platform. Overconstraints in loop h_1 - h_4 - h_5 - h_2 should be released in link b_7 .

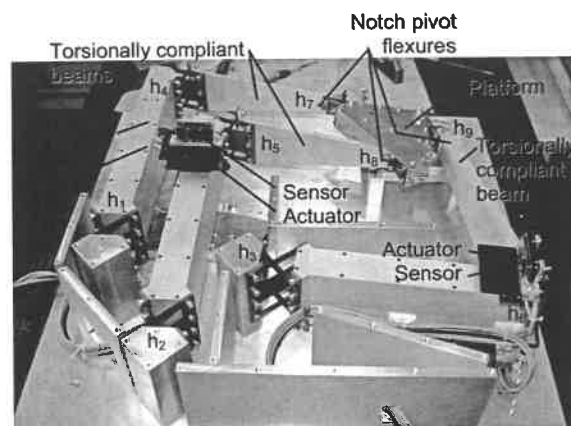


Figure 6. Two-DOF stage. All hinge-joints are made by cross flexure hinges. Hinges h_{10} and h_{11} and the top frame plate are not implemented.

CONCLUSION

Only exact constraint designs show a combination of high stiffness and low internal stress arising from a possible misalignment. The SVD analysis can be conveniently used to choose the best location and direction for implementing release flexures. With the aim of maximizing the stress reduction, release flexures should be implemented in the part of the overconstrained loop where the internal moment is the highest. At the same time the stiffness at the end-effector can be kept high. The design approach using SVD analysis leads to predictable, high stiffness and low internal stress designs.

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DESIGN & FABRICATION OF A PRECISION PULSED VALVE SYSTEM FOR NEUTRON IMAGING

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INTRODUCTION

The development of a high-energy (10Mev) neutron imaging system at Lawrence Livermore National Laboratory (LLNL) depends on a precision engineered pneumatic driven valve system for creating the neutrons that are used for imaging dense objects.

This valve system creates pulses of deuterium gas that interact with a high energy pulsed beam and thereby create neutrons that are used for imaging. This subsystem is part of a larger system which includes a linear accelerator that creates a deuteron beam, a scintillator detector and high resolution CCD cameras.

SYSTEM DESIGN

The mechanical design consists of a hollow titanium double acting piston as shown in Figure 1. This piston is attached to a titanium tube that serves as a bearing inner surface that locates the piston radially as it moves up and down as shown in the complete assembly in Figure 2. The piston is housed inside a titanium cylinder with titanium manifolds at both ends.

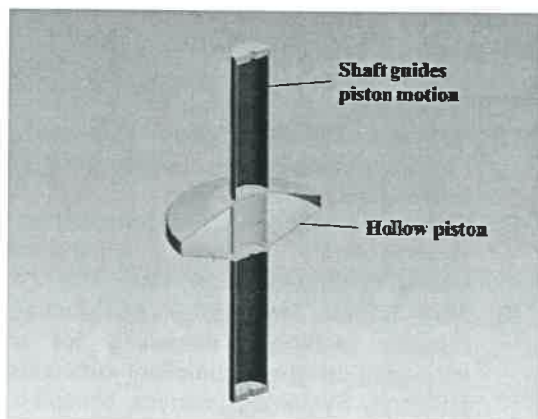


FIGURE 1. Hollow piston cross section.

Timed air pressure at 7 atmospheres enters the manifolds to drive the piston up and down. This in turn drives a poppet valve located in a gas cell

as shown in Figure 3. As the valve opens and closes it charges a gas cell with up to 3 atmospheres of deuterium. At the peak gas cell pressure a pulsed deuteron beam fires through the beam hole to cause an interaction with the deuterium and thereby create neutrons. These neutrons continue on down the beam path for use in imaging.

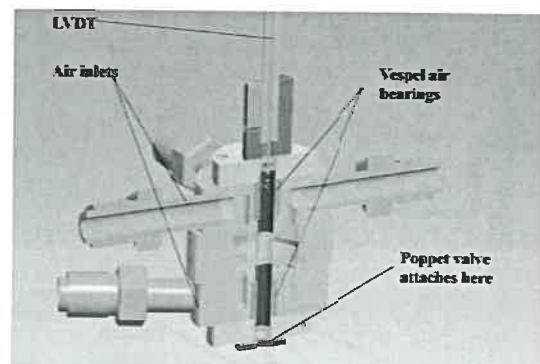


FIGURE 2. Complete piston and cylinder assembly cross section.

The valve opening and closing was designed to operate at a maximum speed of 1000 Hz in order to generate a deuterium pressure pulse to match the repetition rate of the beam pulses. In order to achieve this, the piston assembly was design to be supported laterally on air bearings at both ends where the piston tube passes through the manifolds. The air bearings will be made of vespel and will have stepped circumferential pocket designed to operate at 6 atmospheres. With a 0.5 micron step the radial load capacity was calculated to be approximately 5N (1.12lbf) [1][2]. Thereby maintaining a 1.3 micron radial clearance between the piston and the cylinder wall.

The timed air pressure that drives the piston up and down is controlled by Festo solenoids that are driven by a LabView program through a National Instruments digital-to-analog converter as shown in Figure 4.

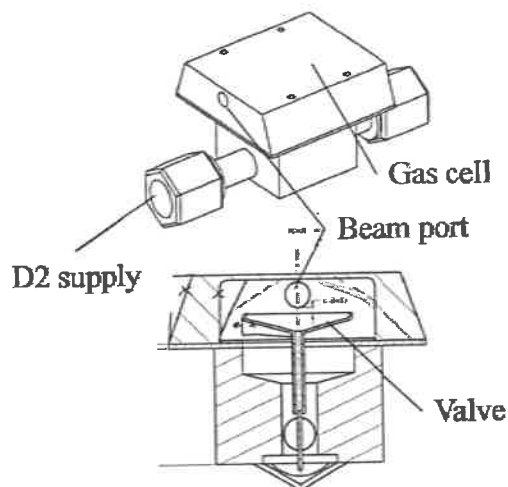


FIGURE 3. Gas cell and cross section

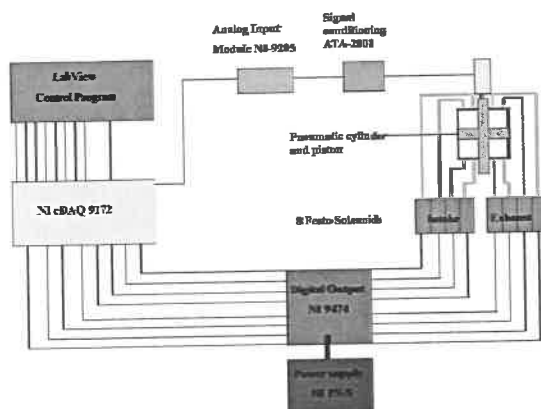


FIGURE 4. Supply and control schematic.

SYSTEM MODELING

Modeling of the pneumatic system was done by using a lumped parameter method (bond graphs) to characterize the mass flow and enthalpy into and out of each chamber as an isentropic nozzle [3]. The state equations shown below define the flow of gas into the piston's chamber as show in Figure 5.

$$\dot{m}_1 = A_1 \frac{P_u}{\sqrt{T_u}} \sqrt{\frac{2\gamma}{R(\gamma-1)}} \sqrt{Pr^{2/\gamma} - Pr^{(\gamma+1)/\gamma}} \quad (1)$$

$$\dot{E}_1 = C_p T_u A_1 \frac{P_u}{\sqrt{T_u}} \sqrt{\frac{2\gamma}{R(\gamma-1)}} \sqrt{Pr^{2/\gamma} - Pr^{(\gamma+1)/\gamma}} \quad (2)$$

Where:

$\dot{m}_1$ = mass flow into chamber 1

E_1 = enthalpy flow into chamber 1

A_1 = flow area

P_u = upstream pressure

P_d = downstream pressure

C_p = specific heat constant pressure

C_v = specific heat constant volume

$$Pr = \frac{P_d}{P_u} \quad \gamma = \frac{C_p}{C_v} \quad R = C_p - C_v$$

The flow area was made time dependent and this was used to simulate the opening and closing of the solenoid intake and exhaust valves. The enthalpy in equation (2) was then used to solve for the pressure in chamber 1, given the volume of chamber 1,

$$P_1 = \frac{RE_1}{C_v V_1} \quad (3)$$

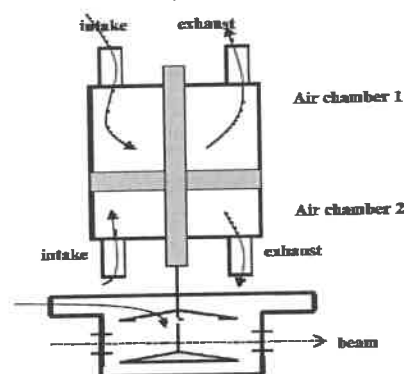


FIGURE 5. Modeling of piston, valve motion and the gas cell.

The differential equations (1) and (2) were numerically integrated in a Matlab simulation for both air chambers 1 and 2. The chamber pressure for both sides of the piston, as a function of time, was solved using equation (3) applied to each chamber. Thus yielding pressures P_1 and P_2 and the net force on the piston as shown below:

$$F_p = (P_1 - P_2) \cdot \text{piston area} \quad (4)$$

This was then used to solve for the motion of the piston:

$$\dot{p}_{\text{piston}} = F_p - m_{\text{piston}} g \quad (5)$$

$$\dot{q}_{\text{piston}} = \frac{p_{\text{piston}}}{m_{\text{piston}}} \quad (6)$$

where: p_{piston} = piston momentum
 m_{piston} = piston mass
 q_{piston} = piston motion

The poppet valve is directly connected to the piston so that the area of the open valve can be calculated as follows:

$$\text{valve open area} = q_{\text{piston}} \cdot \pi \cdot \text{valve diameter} \quad (7)$$

The same approach was used to determine the pressure in the gas cell by taking the valve open area in equation (7) and using it in equations (1) and (2) as the variable intake area for determining the mass and enthalpy flow of deuterium. This was balanced by also using the same equations with different areas to determine the net outflow of deuterium through the beam port holes. The pressure was determined by solving:

$$P_{\text{gas cell}} = \frac{R(E_{\text{valve}} - E_{\text{beam port}})}{C_v V_{\text{gas cell}}} \quad (8)$$

Where:

$P_{\text{gas cell}}$ = Pressure in gas cell

E_{valve} = Enthalpy flowing through the valve

$E_{\text{beam port}}$ = Enthalpy flowing through beam ports

The simulation of the complete system yielded valve motion and deuterium gas cell pressure as a function of time as shown in Figure 6.

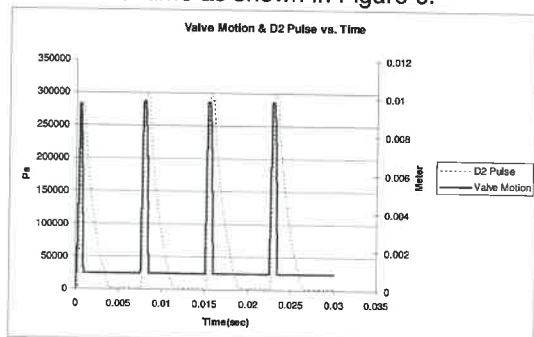


FIGURE 6. Valve motion & deuterium pressure pulse vs. time.

The valve in Figure 6 opens to a maximum displacement of 10mm every 7.5 milliseconds. This leads to a maximum deuterium pressure of 3 atmospheres that lags behind the valve motion by about a millisecond. The beam would be timed to fire at maximum pressure in the gas cell, thus yielding the highest production of neutrons.

In order to achieve the desired deuterium pressure pulse, the valve had to open and close within a millisecond. This would be achieved by timing the opening and closing of the solenoid valves as shown in Figure 7. Through simulation trial and error it was found that the simulation command pulses had to overlap in order to deal with the inertia in the system. Therefore the exhaust solenoid valve command on the top piston would have to open 2.5 milliseconds before the intake solenoid closed. Similarly the intake on the bottom chamber would open 0.5 milliseconds after the opening of the top solenoid. Also it was found in the simulation that the piston could be slowed before reaching its top and bottom position by adjusting the timing of the solenoids on the side where the piston was in compression. Thereby reducing the impact on the piston tube as it changed direction.

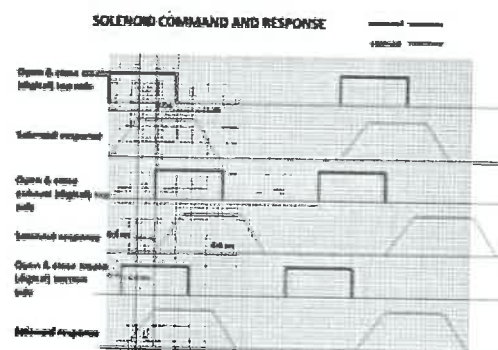


FIGURE 7. Command and solenoid response schematic.

The effect of blow by on cylinder pressure was simulated using equations (1) and (2). The area in the equations was the gap between the piston and the cylinder. The 1.3 micron radial gap was predicted to reduce the peak cylinder pressures by 10%.

FABRICATION

The piston and tube were machined out of titanium (Ti 6Al-4V) in sections. Titanium was chosen because of its high strength, lightness, fatigue properties and its favorable microstructure after laser welding. Figure 8 shows a cross section of the piston and its laser welded components. The top of each side of the piston was angled upward to reduce peak stresses in the structure due to the 7

(coil 1) and two poles of the permanent magnets make Lorentz forces which generate the Θ_x - tilt torques. The other two coils(coil 2) generate the Θ_y - tilt torques. Then tilt torques are generated for driving the Θ_x - tilt and the Θ_y - tilt motions independently.

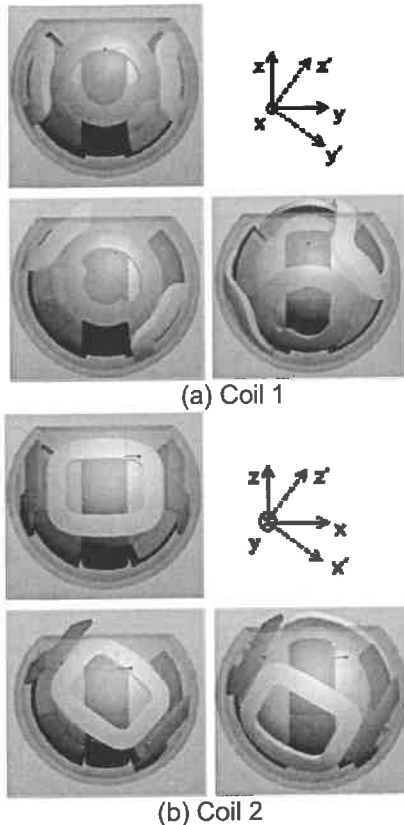


FIGURE 3. Directions of the torque vectors generated by the coil pairs

If the system uses gimbal guide mechanism, guiding directions are same as those tilting torques are exerted. The pair of the Θ_y - tilt coils(coil 2) always generate y-direction torque and another pair of the Θ_x - tilt coils(coil 1) always generate x'-direction. Therefore, two tilt torques are transmitted to gimbal mechanism which guides rotor.

The proposed system uses only VCM principle to generate the tilt motions. For that reason, the structure and the driving principle of the system are simple. Therefore, easy and accurate modeling is possible. Moreover, the torque constants in 2-DOF of the system are always constant. It can be controlled easily because

driving each axis individually in arbitrary position. This system can develop high flux density in air gap. Therefore, high torque constants of tilt and rotation part can be provided within compact size.

SIMULATION

FEM simulation

To confirm this design FEM simulation is carried out. The simulation parameters for FEM feasibility test are shown in the table 1. Maxwell 13 is used for simulation of the magnetic flux and torque analysis.

TABLE 1. Simulation parameters

Air gap	Magnet thickness	Coil diameter	# of coil turns
5 mm	10 mm	0.35 mm	400

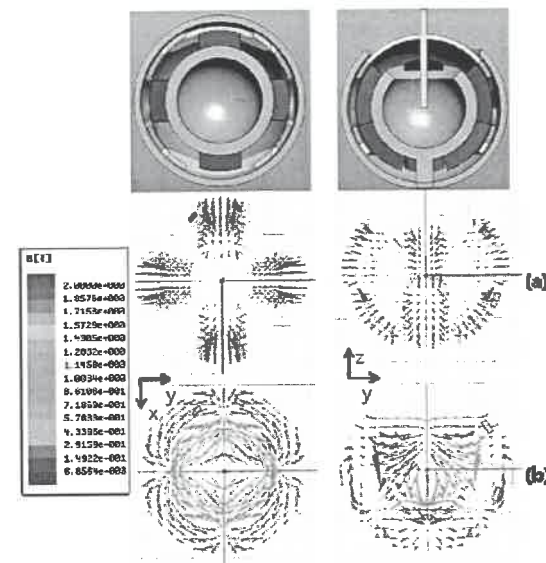


FIGURE 4. Simulation results of the flux density of the system

The figure 4-(a) represents the cross section of the system and flux density formed by permanent magnets in air gap. The figure 4-(b) shows flux density formed by permanent magnets in the yokes. The simulation results of the system are shown in the table 2.

TABLE 2. Simulation results

Magnetic flux density	Torque constant
0.65 T	0.4 Nm/A

The simulation results show the high magnetic flux density in the air gap and the high torque constant within compact size.

CONCLUSION

In this study, a Novel 2-DOF Spherical Motor with the VCM Principle is proposed and simulated. Outer diameter of the stator of the system is only 100 mm. This system can be modified to 3-DOF spherical motor. It just need to add rotational(Θ_z) part. Future works include design of a rotational part for 3-DOF spherical motor, a guide mechanism and optimization of the proposed spherical motor.

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CHARACTERIZATION OF TRANSLATION OF FUSED SILICA MICROPIPETTES IN NON-RECTILINEAR TRAJECTORIES

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INTRODUCTION

In vivo imaging technologies such as confocal endomicroscopy used for clinical histology applications, as well as *in vivo* patch clamping electrodes used in neuroscience research require very precise and stable positioning of probes within a living test subject [1-2]. For these procedures to be as minimally invasive as possible the probes need to navigate through the body's canals and cavities to reach the region of interest and are often guided using flexible and reconfigurable guide tubes in complicated non-rectilinear trajectories. Current technology used for controlling the translation of such probes within the guide tubes do not have the requisite axial resolution (1-2 μm) that will allow imaging or recording of electrophysiological activity from single cells [2]. Understanding the mechanical parameters that affect the performance of such systems will allow the development of micrometer resolution actuation systems.

We have developed an experimental methodology in combination with flexible multibody simulations to study the translation of flexible polyimide coated fused silica micropipettes (typically used for *in vivo* electrophysiology studies) telescoping within Teflon tubing that are configured in rigid predetermined curved configurations. We are investigating the performance of this translation system over a wide spectrum of geometrical (bend radii, arc length, tubing to capillary diameter clearance) and physical (preload,

frictional force) parameters and the results of this study will help in developing a basis for design of more complex non-rectilinear probes.

EXPERIMENTAL SETUP

Shown in Figure 1 is the experimental setup being used to study the geometrical parameters that affect the translation of fused silica capillaries. It consists of: a) linear actuation system comprising a piezomotor and linear stage capacitive sensing system. The fused silica micropipettes (658 μm outer diameter) are attached to the linear stage for actuation using a micropipette holder and are routed through Teflon guide tubes (21 gauge, or 724 μm inner diameter) that are fixed in predefined circular grooves in an acrylic plate wedged between two aluminum plates. The aluminum plates have holes corresponding to the grooves where the Teflon guide tubes terminate and allow the normal entry and exit of the micropipettes. An aluminum disc is attached via a precision-machined shaft translating in a linear roller bearing to the end of the micropipette acts as the ground plate for the capacitive position sensor. Thus actuation of the micropipette by the piezomotor results in corresponding axial displacement of the ground plate that can be measured.

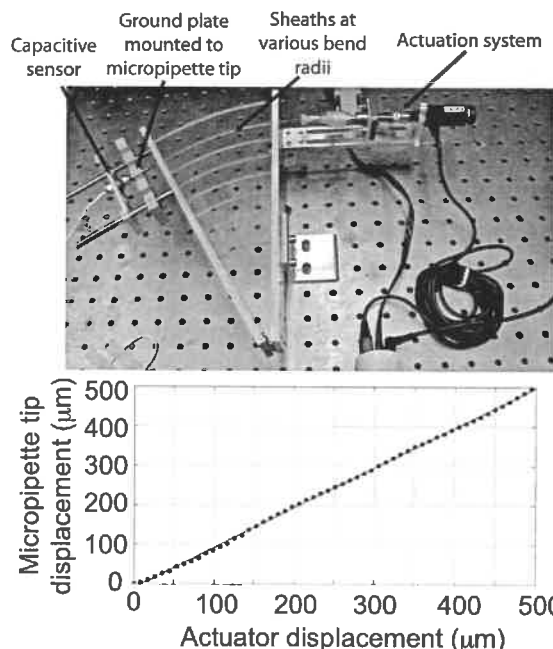


FIGURE 1: (top) Photograph of experimental setup used for characterizing micropipette translation. (bottom) Representative plot of micropipette translation through a Teflon guide tube configured at 160 mm bend radius and 45° arc angle ($r^2=0.987$).

This setup allows characterization of the linearity of micropipette translation through guide tubes of radii ranging from 80, 100, 120, 140, 160, 180, 200 and 220 mm respectively and spanning arc angles of 45°. Shown in Figure 1b is a representative plot of translation of a micropipette through a Teflon guide tube having a 160 mm bend radius and spanning an arc angle of 45° at step input of 10 μm . We were able to measure good linearity in the micropipette tip displacement ($r^2=0.987$) in all bend radii configurations.

In a second set of experiments, we measured the hysteresis in translation of the micropipette and in various geometrical configurations during bi-direction motion. The results are shown in Table 2 and accompanying Figure 2. The input displacement of the micropipette is 10 steps of 50 μm forward and 10 steps back. As it can be seen, there is an inverse linear correlation between the measured hysteresis and the bend radius, that is, the hysteresis is largest for the smallest bend radius for the same spanned arc length. The quantification of this relationship between hysteresis and geometric configuration will guide the design of future devices that required bi-directional non-rectilinear translation.

TABLE 1: Measured hysteresis for 658 μm diameter micropipette translating inside of 21 gauge (724 μm) Teflon tubes through an arc of 45°.

Bend Radius	Hysteresis
80 mm	45 μm
120 mm	-
160 mm	33 μm
200 mm	26 μm

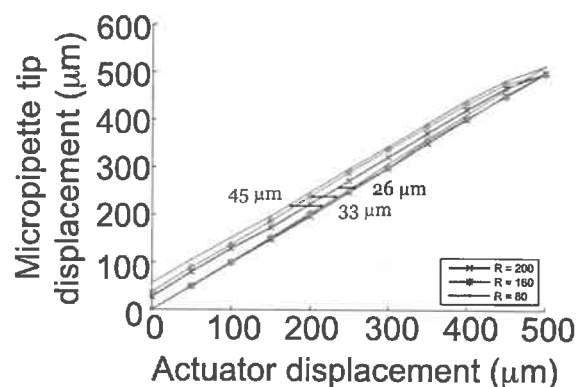


FIGURE 2: Measured hysteresis in displacement at various bend radii. The results show an inverse correlation between bend radius and hysteresis.

MULTIBODY SIMULATIONS

A multibody system analysis approach based on non-linear finite elements using SPACAR [3] is used to model the micropipette [4]. The multibody approach allows for modeling with a limited number of elements whilst obtaining accurate results. A micropipette is modeled as a series of interconnected flexible beam elements. Each beam has six deformation modes: axial, torsion and four bending deformation modes. The beams are connected at nodal points. At the nodal points external forces such as the normal reaction and friction forces can be applied. The guide tube is modeled as a tube of uniform circular cross-section with flexible walls with centerline tracing a circular arc. The contact force between the micropipette and the guide tube depends on the distance and rate of approach between them. Three regions are used; a no contact region, a full contact region, and a transition region in between to make the model continuous. In the full contact region the interaction between the micropipette and the wall results in a normal force that depends linearly on the displacement of the micropipette

wall interface, that is, a constant stiffness regime. In addition, a velocity dependent damping force is added. With the normal nodal force and tangential velocity known, a Stribeck based continuous friction model is implemented. The friction is increased at low velocity to represent static friction. The friction force always acts opposite to the direction of motion on the tangential plane at the point of contact. Simulations can be carried out both for the insertion of the micropipette as well as for the fine positioning of the tip. During a simulation the nodal displacement and force acting at the various nodes have been obtained depending on position and time.

Figure 3 shows the forces acting at the first two nodes at the distal end of the micropipette when it is inserted into the guide tube, neglecting friction. Only the first two nodes are considered for the sake of clarity, though similar plots can be obtained for all nodes. It can be observed that the end node is making contact with the outer wall of the tube all the time, whereas the penultimate node is making contact with the inner wall as expected. Clearly the nodes experience different forces. Figure 4 shows the magnitude of forces acting at the first two distal nodes in time. The penultimate node experiences a larger force in the beginning when the end node started making contact with the tube.

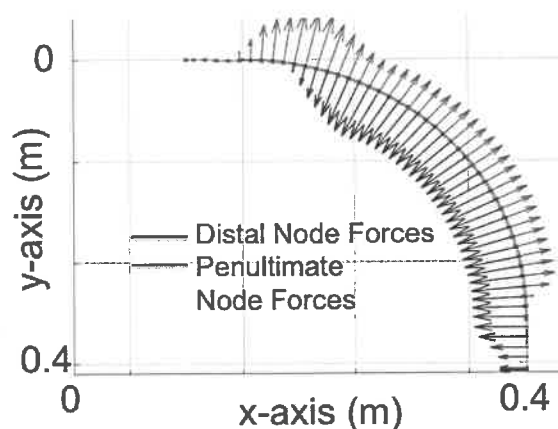


FIGURE 3: Forces as a function of space at the two distal nodes while inserting micropipette translating into tubes through an arc of 45° .

As the micropipette advances further into the tube, the first two distal nodes experience larger forces than the other nodes.

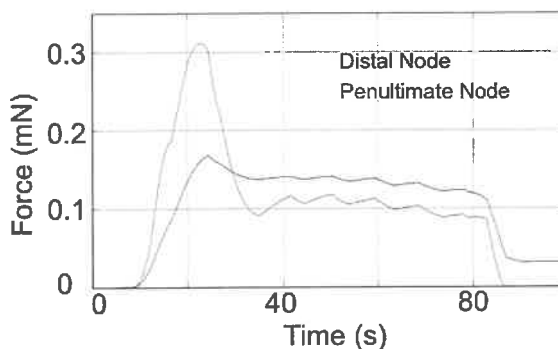


FIGURE 4: Forces as a function of time at the two distal nodes while inserting micropipette translating into tubes through an arc of 45° .

A similar input displacement to the experiments presented earlier at a slightly larger velocity is used in the simulations. The results for three simulations are shown Table 2 and Figure 5. Two simulations are performed with a friction coefficient $\mu = 0.02$ and one with a friction coefficient of $\mu = 0.2$. This range of the friction coefficient can be found for Teflon on Teflon contact. The bend radius is $R = 80$ mm and $R = 120$ mm.

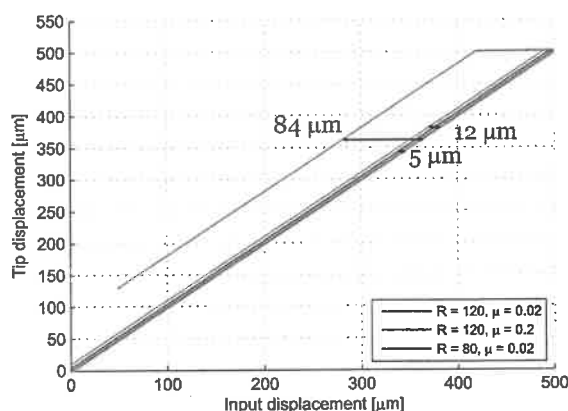


FIGURE 5: Simulated hysteresis in displacement at various bend radii.

The simulations show a larger hysteresis for a smaller bend radius, which is in agreement with the experiments. The nodal forces increase due to a decreased bend radius of the micropipette, which causes friction and therefore hysteresis to increase. The hysteresis also increases for

larger friction coefficients. The simulated hysteresis ranges from 5-84 mm for $R = 120$ mm which is due to the large spread in friction coefficient and large radius. The experimental data for $R = 120$ mm is not available. The experimental hysteresis of $R = 80$ mm and $R = 160$ mm is similar to the simulation results for $R = 120$ mm. The experimental values, with a measured friction coefficient of about 0.10 to 0.14, fit within the results of the simulation.

TABLE 2: Simulated hysteresis for 658 μm diameter micropipette translating inside of 21 gauge (724 μm) Teflon tubes through an arc of 45° .

Bend Radius (mm)	Friction Coefficient	Hysteresis (μm)
80	0.02	12
120	0.02	5
120	0.2	84

CONCLUSIONS

The experiments conducted show that it is possible to achieve highly precise translation of flexible micropipettes in defined non-rectilinear trajectories. We have shown that by varying the bend radius systematically, we can quantify the relationship between hysteresis and geometrical configuration. Both experimentally and through simulations, we have shown an inverse linear correlation between the measured hysteresis and the bend radius, that is, the hysteresis is largest for the smallest bend radius for the same spanned arc length. In the future more simulations will be set up to verify the experimental data with more accurate radii and friction coefficients. Currently the simulations are slow and very sensitive to input parameters such as the friction coefficient. More superficial models are being worked on using a state friction models or other integrators. This combined approach of experimental measurement and multibody simulations can be extended to screen a wide range of geometrical configurations and in future be used to accurately predict the translation of such tubes in arbitrary curved configurations.

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DESIGN OF ARTIFICIAL HIP JOINT SIMULATOR FOR WEAR TEST

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INSTRUCTIONS

The human's hip joint can be damaged by injury or various arthritis, and the mankind has developed total hip arthroplasty to eliminate the damaged hip joint and to insert artificial materials to replace the hip joint. A modern total hip arthroplasty has been started through Themistocle Glück on 1890s. Glück invented an artificial hip joint that was made of ivory, using synthetic resins, stone, or plaster composites. After that, Wiles designed artificial hip joint with stainless steel in 1938. On the end of 1950s, a high molecular substance which was introduced by Charnley helped to reach an advanced progression in the field of artificial hip joint. Recently, an ultra-high-molecular-weight polyethylene and ceramic are developed, and they are used to design an artificial hip joint. [1]

Along with a progression of artificial hip joint replacement, research about the impact of implanted artificial hip joint towards the human body also has been progressed. The clinical results of artificial hip joint's impact towards human body are like below. The first generation of artificial hip joint: High molecular substance (polyethylene) wears off and the abrasion particles melt the bones near by the artificial joints which cause the problem replacing it again. The abrasion of a cross-linked ultra-high-molecular-weight polyethylene substance, γ -rayed polyethylene substance, is decreased 80%, approximately; however osteolysis, by created abrasion particles, is still occurred and leaves a problem. The second generation: metal substance (Co-Cr alloys) is not sufficient enough because it increases the ion density of the body fluid, or the abrasion particles fall off and cause the allergy. The third generation: ceramic material decreased in abrasion from the previous substances' problem; however the

artificial joint can be break easily due to its fragility. [2]

Ceramic-ceramic bonded artificial joint is widely used in recent artificial joint replacement procedures. This means that the abrasion causes more problem than ceramic's fragility in artificial joint replacement procedure. Although the ceramic-ceramic bonded joint is largely used, it's hard to be applied on young-aged generation because the fragility of ceramic will not be able to support young people's massive activities. Therefore, it is necessary to research to solve abrasion problems such as surface-treating the previous materials, or to revise the shape of the materials along with developing a new substance that has similar mechanical characteristic with human bone and that performs similar to human bone so that the artificial joint can be applied in various age group. To satisfy the previous necessities, developing the machine which can evaluate the abrasion characteristic of artificial joint is also inevitable.

Right now, there are some applied testing simulator for the measurement and evaluation of abrasion using self standards [3]; however, in this lab, an abrasion testing simulator that can manipulate angular motion of artificial hip joint is designed to measure the abrasion of artificial hip joint, followed by industry standards ISO 14242-1. The reason why this lab followed ISO 14242-1 is because the ISO/FDIS 14242-1 is accepted in general, and because the artificial joint abrasion test standards is highly required. In this lab, other factors such as the weight difference depends on time period, and lubrication system also will be simulated on designed abrasion testing simulator.

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ISO STANDARD OF MEASUREMENT

The weight, displacement variables, and testing environment about artificial hip joint abrasion testing simulator are standardized by industry standard ISO 14242-1. According to the standard, testing simulator should be able to create a specified angular displacement as it shows on Figure 2. Specified angular displacements are as follow; 25°, -18° in flexion/extension, 3°, 7°, -4° in abduction/adduction, and -10°, 2° in inward/outward rotation angle. The motion control system of the testing simulator should be able to create angular motion of components with accuracy of $\pm 3^\circ$, at the least and at the most, when it is asked to do three different performances, along with $\pm 1\%$ of cyclic time period for the phase change. [4, 5]

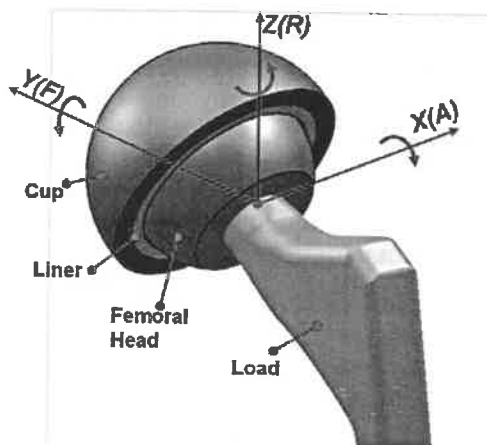


Figure 1. Schematic view of hip joints.

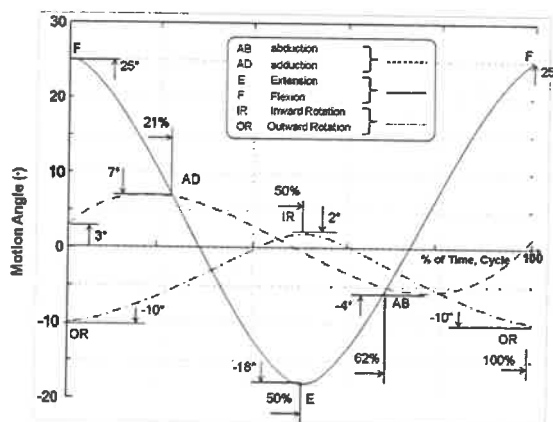


Figure 2. Time vs. displacement graph of angular motion that is applied to femoral region specimen.

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TABLE 1. Displacement of angular motion that is applied to femoral region specimen depends on the time period.

% of time, cycle $\pm 1\%$	0	21	50	62	100
Flexion(+) or extension(-) $\pm 3^\circ$	25		-18		25
Adduction(+) or abduction(-) $\pm 3^\circ$	3	7		-4	3
Inward(+) or outward rotation angle $\pm 3^\circ$	-10		2		-10

STRUCTURAL CONFIGURATION

Hip joint motion can be express with analysis of 3D system as Figure 2 shows, and the testing simulator is designed with double arch-type, 3-axis variable simulator, like Figure 3, to manipulate the hip joint motion. A remarkable point of this double arch-type, 3-axis variable simulator is that this testing simulator can apply rotation motion which is caused by crossing two vertical circular columns. As Figure 3 shows, each of two vertical columns are drove as motors, and upper side supporting area is supported by linear transport motor and cross roller guide. Platform that is supporting spindles manipulate the rotation motion so that the motion can be smooth. Also, the bottom side is able to manipulate three degree of freedom because it contains a single axis rotation device.

Designing a double arch-type, 3-axis variable simulator is held like below. The vertical columns are designed into circular columns to perform count balance and to raise accuracy of the device. Also, a hole is made in the circular

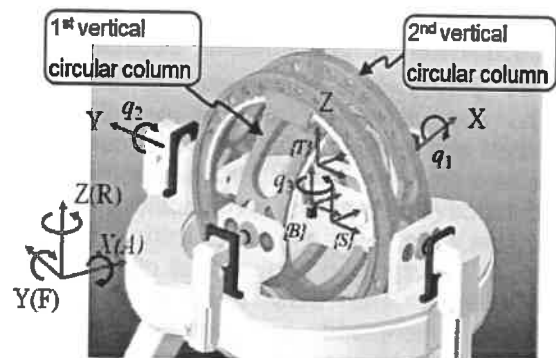


Figure 3. Design of Simulator and Coordinates.

columns to decrease the capacity of motor. In this lab, the analysis of the machine is focused and the coordinate is defined first. The coordinates of this double arch-type machine tool are as follow. The base frame of lower gripper part is {B}, the coordinate of the station on {S}, and the position and rotation of upper gripper is {T}. The rotation matrix is as follow.

$${}^B_T = {}^B_S {}^S_T {}^S_T^{-1} {}^S_T \quad (1)$$

The inverse kinematics is calculated by this relationship between the rotation of the upper gripper and the rotation of the motors. The rotation angle between the upper gripper and lower gripper is calculated first to solve this problem.

KINEMATIC ANALYSIS

The rotation angle α can be calculated by the norm vector of X-axis of the 1st vertical circular

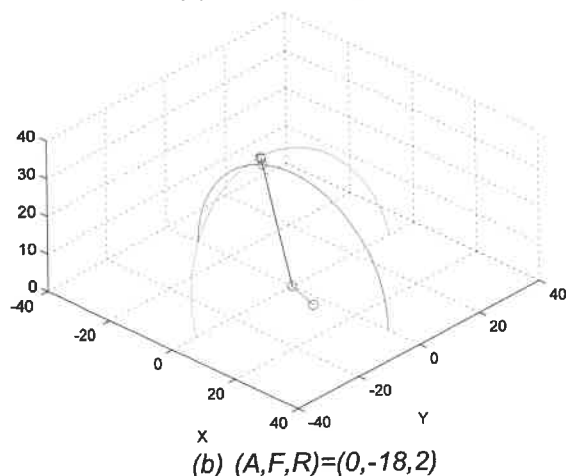
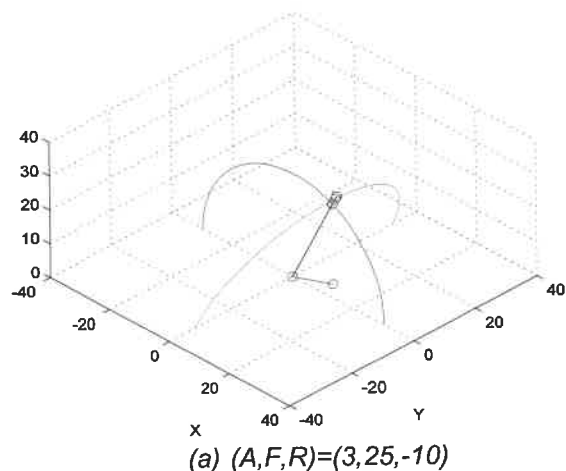


Figure 4. Schematic diagram of simulator.

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column. The vertical vector is laid on XY-plane and the components are $(0, \cos q_1, \sin q_1)$. The plane equation of this vector as a normal vector is as below. The point $(0, 0, 0)$ is on this plane, so $d_1=0$. This is 1st plane and the equation is as follow.

$$(\cos q_1)y + (\sin q_1)z = 0 \quad (2)$$

The rotation angle β can be calculated by the norm vector of Y-axis of the 2nd vertical circular column and the norm vector is $X_2(\cos q_2, 0, -\sin q_2)$. This 2nd plane equation is as follow.

$$(\cos q_2)x - (\sin q_2)z = 0 \quad (3)$$

The intersection of 1st plane and 2nd plane is as shown as straight line. The directional vector of this line is as follow.

$$\text{directed cosine}(Z_2) = \frac{1}{N}(\tan q_2, -\tan q_1, 1) \quad (4)$$

$$N = \sqrt{\tan^2 q_1 + \tan^2 q_2 + 1}$$

Y axis of 2nd plane (Y_2) can be calculated by cross product of X_2 and Z_2 .

$$Y_2 = \frac{1}{N} \left(\tan q_2 \cdot \sin q_2, \frac{1}{\cos q_2}, \tan q_1 \cdot \cos q_2 \right) \quad (5)$$

q_1 and q_2 are the rotation angle of fixed X and Y axis in global coordinates.

$${}^B_T = R_{xyz} = \text{Rot}(\hat{X}, A) \text{Rot}(\hat{Y}, F) \text{Rot}(\hat{Z}, R) \quad (6)$$

$${}^S_T^{-1} = \text{Rot}(\hat{Z}, q_3)$$

$${}^S_T = [X_2 \ ; \ Y_2 \ ; \ Z_2]$$

The inverse kinematic solution is as follows.

$$\begin{aligned} q_2 &= -\text{asin}\{R_{xyz}(3,1)\} \\ q_3 &= -\text{asin}\left\{\frac{R_{xyz}(2,1)}{\cos(q_2)}\right\} \\ q_1 &= \text{atan}\left\{\frac{R_{xyz}(3,2)}{\cos(q_2) \cdot R_{xyz}(3,3)}\right\} \end{aligned} \quad (7)$$

MOTION ANALYSIS

Base on this inverse kinematics formula above, mobility will be analyzed. First, some locations

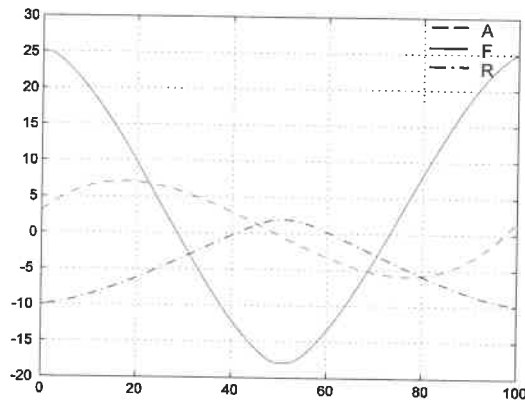


Figure 5. Input values(AFR) of the rotation.

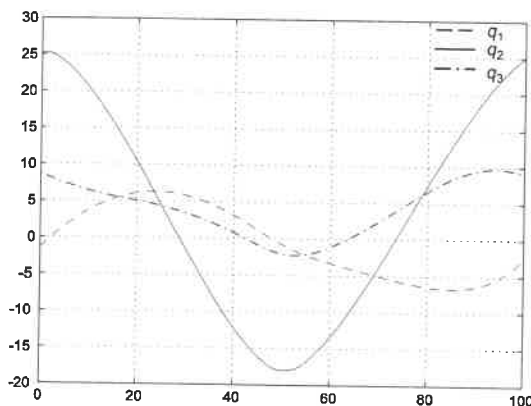


Figure 6. Encoder value(q_1 , q_2 , q_3).

are schematized in Figure 5 and Figure 6, using value of the inverse kinematics formula. In the simulation, rotation direction of vertical columns is matched as the angle degree changes. Likewise, lower gripper part performs relative motion since it moves to opposite direction in inward/outward rotation (R) section; the R value is -10, however the lower gripper part rotates in about 10 degrees in Figure 5. Therefore, relative motion is completed.

Through this analysis, the possibility of hip joint abrasion test on this simulator can be evaluated. Followed by ISO 14242-1 standard, the rotation is defined; flexion/extension (F) value has a range of -18° ~ 25° , abduction/adduction (A) value has -4° ~ 7° , and inward/outward rotation (R) value has -10° ~ 2° . The each value of encoder using the inverse kinematics formula is like next; q_1 exists in a range of -6.8° ~ 6.3° , q_2 exists in a range of -18.0° ~ 25.3° , and q_3 exists in a range of -2.2° ~ 9.8° . q_1 , q_2 are the work domain of apparatus that is occurred by the interference, and they represent -60° ~ 60° .

As the analysis from above shows, this simulator from the lab satisfies the motion range that is defined in ISO 14242-1 standard, and it also satisfies the specification as a simulator. Furthermore, this simulator from the lab fits sufficiently on performing various motions because it has larger work domain.

CONCLUSION

In this research, double arched, 3-axis variable simulator is designed to test the abrasion rate between head of femur and acetabular cup of artificial hip joint. To follow the industry standard ISO 14242-1, this testing simulator is designed to create a accuracy of $\pm 3^{\circ}$ at the least and at the most, and to create an angular motion of femoral component (flexion/extension, abduction/adduction, inward/outward rotation) along with the cyclic time period of $\pm 1\%$ for the phase change. The design theory is based on inverse kinematics, and is simulated through MATLAB to confirm whether the motion range of the abrasion testing simulator fits in ISO 14242-1 or not. As result, it is confirmed that the apparatus' working territory which is caused by the interference is measured in a range of -60° ~ 60° ; thus, the testing simulator covers the motion range which is defined in industry standard sufficiently. Large working domain will allow performing various motions. In this research, weight difference along with the time period, and lubrication system will be also researched after designing the abrasion testing simulator.

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Development of a Novel Compact 3-DOF Out-of-plane Stage

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INTRODUCTION

A semiconductor industry is an important part of the whole industry and has become more and more developed. According to development, the manufacturing process of a semiconductor element has become more various and requires high technology. Also, high precision technology needs a precision manufacturing machine. So, these processes need a various high precision stage.

There are many kinds of actuators to achieve high precision. Voice coil motor (VCM) is one of them. But, VCM has disadvantages such as thermal problem and coil alignment problem. And ball screw is another actuator that has a disadvantage of backlash. But, a monolithic flexure hinge mechanism is the most useful method to achieve high precision and has no thermal problem and backlash.

Previous works of 3-DOF out-of-plane stage didn't obtain both compact size and high natural frequency.[1,2] In this paper, two flexure hinge mechanisms are described with modeling and FEM simulation. These two flexure hinge mechanisms acquired compact size and high natural frequency using a new amplification mechanism.

DESIGN AND OPTIMIZATION OF THE FLEXURE STAGE

Conceptual design

The main purpose of this proposed stage is the wafer alignment stage. Wafer stages require high natural frequency and compact size. This paper proposed a new bridge amplification structure. System basic specifications are represented in Table 1.

Table 1. Stage basic specification

Stage Size	150X150X30
Maximum displacement	>100 μ m(Z)
Maximum rotation angle	> ± 1 mrad ($\theta_{x,y}$)

Design of Amplification mechanisms

The proposed stage contains four amplification mechanisms. Each amplification mechanism contains a PZT actuator.

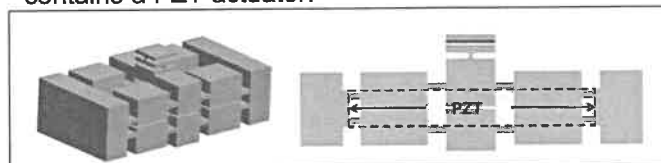


Figure 1. Structure of Double bridge amplification mechanism.

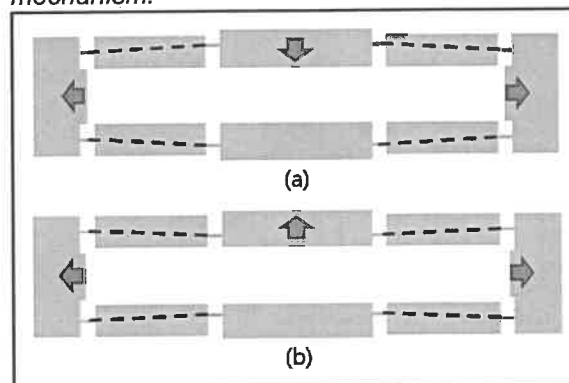


Figure 2. Amplification mechanism

Double Bridge Amplification Mechanism

Fig. 1 shows the Double bridge amplification mechanism. The double bridge amplification mechanism consists of two bridge structures. And a piezoelectric actuator is located between the two bridge structures. Therefore, the two bridge structures can reduce the height of the stage. Fig. 2 represents two kinds of amplification mechanisms which are composed of eight leaf springs. The proposed stage uses mechanism (b).

Design of out-of-plane stage

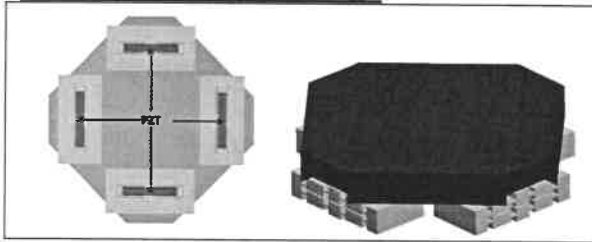


Figure3. Structure of out-of-plane stage

The proposed stage contains four double bridge amplification mechanisms. Each amplification mechanism located center side of square. This arrangement of amplification mechanism can produce a hollowness type stage. Also the arrangement is easy to control.

MODELING AND OPTIMIZATION

Modeling

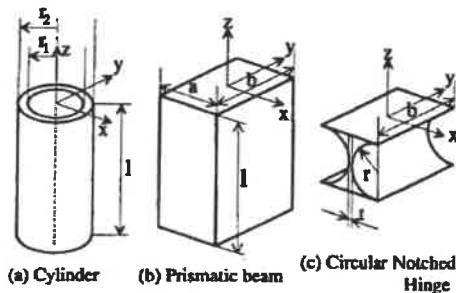


Figure4. Beam and circular hinge coordinate systems

The flexure mechanism of stage consists of rigid links and flexure hinges. Fig. 4 shows the coordinate systems of the leaf-spring and the circular hinge.

$$C^h = \begin{bmatrix} \frac{\Delta x}{F_z} & 0 & 0 & 0 & 0 & 0 \\ 0 & \frac{\Delta y}{F_y} & 0 & 0 & 0 & \frac{\Delta y}{M_z} \\ 0 & 0 & \frac{\Delta z}{F_z} & 0 & \frac{\Delta z}{M_y} & 0 \\ 0 & 0 & 0 & \frac{\Delta \alpha}{M_x} & 0 & 0 \\ 0 & 0 & \frac{\Delta \beta}{F_z} & 0 & \frac{\Delta \beta}{M_x} & 0 \\ 0 & \frac{\Delta \gamma}{F_y} & 0 & 0 & 0 & \frac{\Delta \gamma}{M_z} \end{bmatrix}$$

The compliance matrices of leaf-spring and circular hinge can be derived respectively from beam theory.[4] Relationships between rigid body and hinges are modeled mathematically using Ryu's algorithm.[3]

Optimization

The objective of the optimization is to maximize the system bandwidth to improve the dynamic characteristics.

$$\text{Minimize } f(l_1, t_1, r_2, t_2, r_3, t_3, th) = \frac{1}{\omega_n^2}$$

Also, there are constraints which consider maximum displacement.

$$Z\text{-axis Maximum displacement} > 170 \mu\text{m}$$

$$\theta_{x,y} \text{ Maximum rotation range} > \pm 1.2 \text{ mrad}$$

Prevention contortion amplification mechanism when stage acts $\theta_{x,y}$ axis

$$\theta_{amp_z} - 0.00005 \leq 0$$

To find the solution, a sequential quadratic programming (SQP) method and MATLAB were used. Following figure shows results of optimal design using MATLAB.

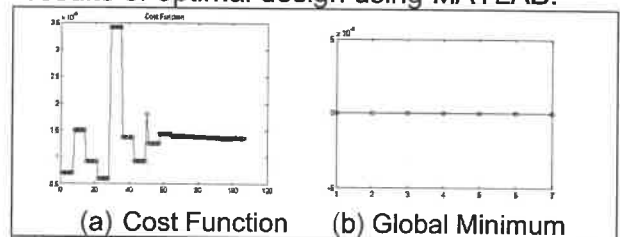


Figure5. Cost function of optimal design

We can confirm results of optimal design reliable.

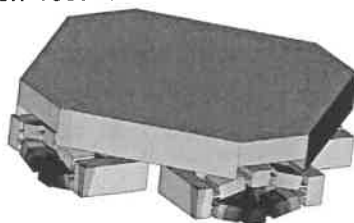
Table 2. Optimal design results

Design variables (mm)	Optimal Design	Final dimension
l	2.996mm	3mm
t	0.974mm	1mm
r	0.517mm	0.5mm
t_1	0.469mm	0.5mm
gap	1.564mm	1.5mm

Table 2. shows results of optimal design. Final dimensions are determined considering minimum tolerance of manufacturing. To verify these analytical results, FEM simulation is performed according to final dimensions.

FEM SIMULATION

Fig. 6 shows analytic results and FEM simulation results.



	FEM	Modelling	Error
Maximum displacement	175.2 μm	164.1 μm	6.33 %
θ_x, θ_y axis Frequency	283.7 Hz	282.21 Hz	0.53 %
Z axis Frequency	268.61 Hz	280.35 Hz	4.37 %

Figure6. Analytic result and FEM simulation results of the Double bridge amplification 3-DOF Stage.

From FEM simulation, we confirmed error around 5%. Therefore we verified accomplishment of modeling and optimal design.

CONCLUSION

In this study, a novel 3-DOF out-of-plane stage is proposed and optimized. The stage attain 100 μm in Z direction and $\pm 1\text{ mrad}$ in $\theta_{x,y}$ direction. Size of the stage is $150 \times 150 \times 30\text{ mm}^3$. Future work includes the fabrication of this stage with final dimension and experiments with the manufactured stage.

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MANUALLY OPERATED PRECISION ASSEMBLY MACHINE FOR BUILDING HIGH POWER LASER TARGETS

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KEYWORDS

fusion, laser target, target assembly, NIF,
National Ignition Facility, Omega Laser

INTRODUCTION

The Non-Ignition Target Assembly Machine (NITAM) was developed to manufacture high power laser targets for the National Ignition Facility (NIF) [1] and the Omega Laser Facility [2]. These targets are different from but related to the laser-driven fusion ignition targets for the National Ignition Campaign [3] that are being built with other machines developed at the Lawrence Livermore National Laboratory (LLNL) [4,5]. The NITAM is a partner machine to a metrology instrument that was recently developed at LLNL for inspecting targets and is presented at this conference: the Non-Ignition Target Inspection Machine (NITIM). Referred to in this paper as the assembly machine, the NITAM was originally planned as an upgrade of certain sub-systems in the legacy assembly machines being used to build the non-ignition targets. Those machines did not accommodate the larger targets that were going to be built, and the low accuracy visual-based displacement feedback used in them – scales and protractors – was limiting the accuracy of the completed targets. The large motion stages in the new machine have optical encoders with digital readouts to provide the needed travel range and accuracy. Two major aspects of the legacy machines were retained: how the operator interacts with it, and the architecture of the motions provided by the manipulator systems.

DESCRIPTION OF THE MACHINE

The assembly machine is depicted in Figure 1. The large motions provided by the manipulator

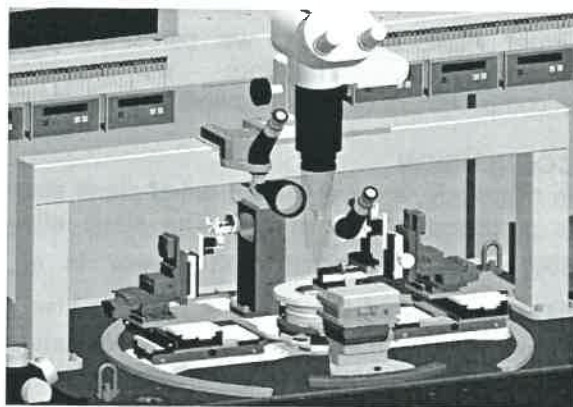


FIGURE 1. Model of the assembly machine.

systems involve eight combined translation and rotary stages. The legacy assembly machines utilize a motion system that is predominantly manually driven, and use a pair of orthogonal microscopes for viewing the target assembly arena. As with the legacy machines, the large motions in the new machine are manually driven instead of being motorized. This satisfied a goal of preserving the intuitive feel that had been acquired during two decades of building targets with the legacy machines. Retaining the familiar use of microscopes with reticles for viewing the assembly arena also supported that goal. The base structure in the legacy machine was restricting the redesign effort and not clean room compatible, so the new machines were designed from the ground up.

BASIC OPERATION

Figure 2 provides a closer view of the assembly machine's manipulator systems. A target base is mounted to the left-side manipulator system,

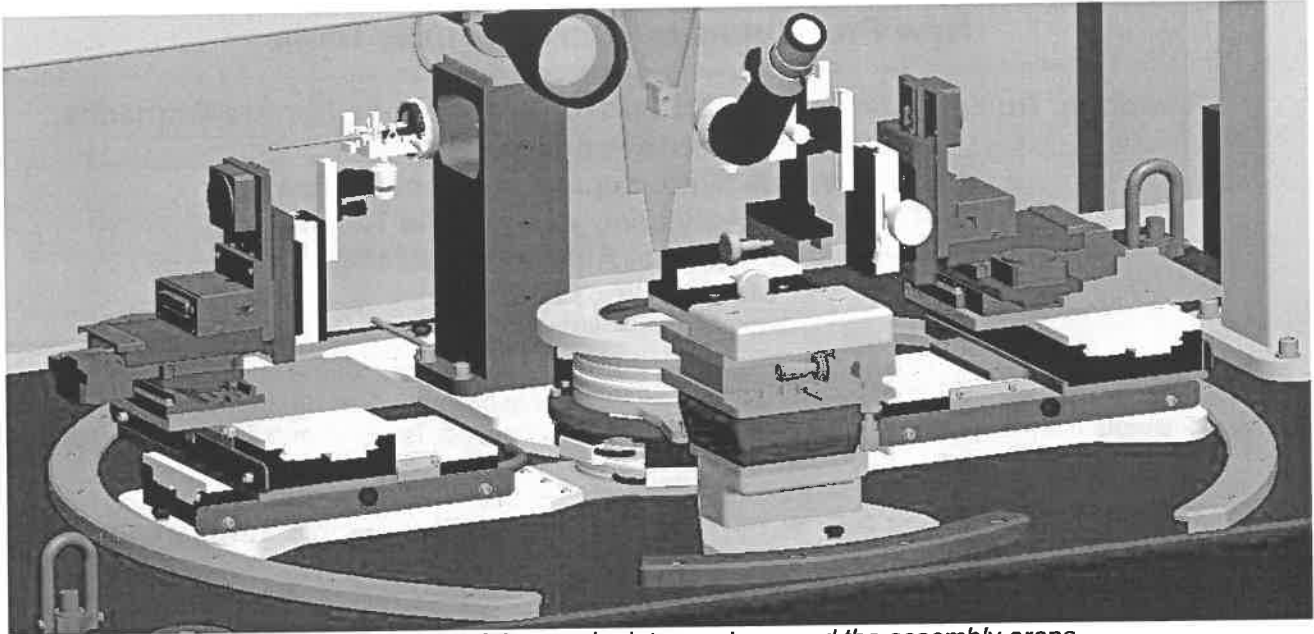


FIGURE 2. Model showing a close-up of the manipulator systems and the assembly arena.

and the target component to be assembled to the base is mounted to the right-side manipulator. These manipulators are carried by independent swing-arms that pivot about a vertical axis aligned to the optical axis of the main microscope. Each of these manipulator systems consist of a pair of orthogonal long-travel translation stages carrying a short-travel motorized XYZ manipulator. An auxiliary vertical stage carrying pitch/roll/yaw tooling is used to attach the target base and target component to their respective XYZ manipulators. The target base manipulator also provides a roll adjustment of the target base around its long axis, which lies in a horizontal plane and nominally crosses the optical axis of the main microscope. Using the right and left swing-arms and their manipulator systems, target components can be aligned relative to the target base in a spherical coordinate system centered at a prescribed location on the target base. A second microscope is mounted to a swing-arm at the front of the machine and provides a horizontal plane view of the assembly arena. A back-lighting system for that microscope is mounted to an independently movable rear swing-arm. A bottom-lighting system provides illumination for the main microscope through an aperture in the central pivot for the swing-arms. All four swing-arms and the four long-travel translation stages have optical encoder feedback that is presented to the operator via the digital displays just above the bridge structure that supports the main microscope.

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New Prototypes of Micromachine Tools

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ABSTRACT

This paper shows the advances in the development of MicroMachine Tools (MMT) in Mexico. Since 1999 the Group of Micromechanics and Mechatronics (GMM) has been testing configurations in order to improve resolution and achieve better precision characteristics. Two novel prototypes are shown. One, based on a Bridge-type configuration, and the other consists on a parallel mechanism arrangement.

BACKGROUND

Since 1999, the development process of the MMT on the GMM involved the fabrication of the whole system, including the actuators, the mechanisms, the structure and the control system. However, during the manufacturing process while using the MMT, many mechanical errors were detected on these prototypes, mainly due to the imprecise mechanical components used [1].

The accuracy of a machine tool is primarily affected by the geometric errors caused by mechanical geometrical imperfections, misalignments and wear of the elements of the machine structure, by the non-uniform thermal expansion of the machine structure and static/dynamic load induced errors. The errors can be reduced with the structural improvement of the machine tool through better design and manufacturing practices. However, in most cases, due to physical limitations, production and design techniques can not solely improve the machine tool accuracy. Therefore, identification, characterization and compensation of these error sources are necessary to improve machine tool accuracy cost-effectively [2].

For that reason, an evaluation method was proposed to improve the characteristics of the MMT prototypes. This evaluation was divided in two analyses: one focused on finding and

solving the geometrical error sources of each mechanism involved on the motion system, and the second, focused on the interaction between those mechanisms.

BRIDGE-TYPE MICROMACHINE TOOL (2010-2011)

Different mechanisms and motion systems were evaluated to determine the best elements to be implemented. The figure 1 shows the selected mechanism. It consists on a commercial stepper motor (200 steps per revolution) first joined to a gearbox reduction (16:1) and after to a leadscrew (M2.5x0.45). The nut moves over two round bar guides, and it has an anti-backlash system using an helicoidal spring.

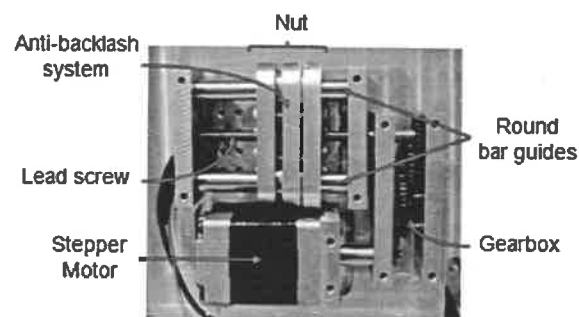


Figure 1. Motion system implemented on the MMT.

A bridge-type configuration was selected for this MMT prototype. According to [3] and [4], closed-loop configurations present more rigidity at the tool tip and a more homogeneous mechanical and thermal behavior than an open-loop configuration. The figure 2 shows a scheme of the MMT prototype showing the closed-loop configuration implemented on the machine.

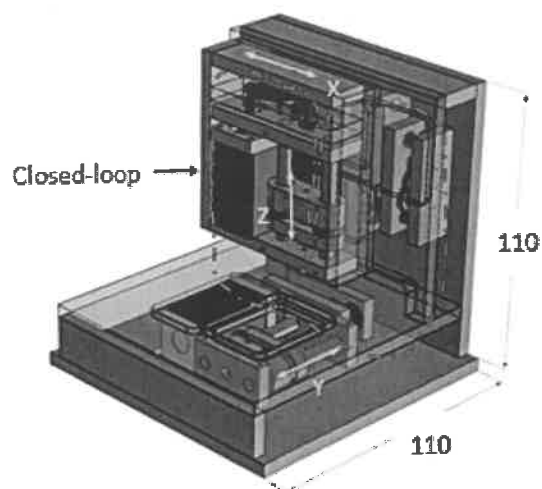


Figure 2. Bridge-type configuration implemented for the MMT developed on 2010.

Finally, an MMT prototype is shown in the figure 3. The characteristics of the system are:

- Dimensions: 110 x 110 x 110 mm³
- Work volume: 20 x 20 x 20 mm³
- Resolution: 140 nm per full motor step
- Max. feed rate: 120 mm/min

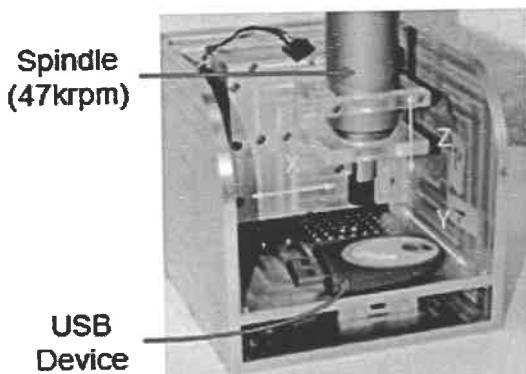
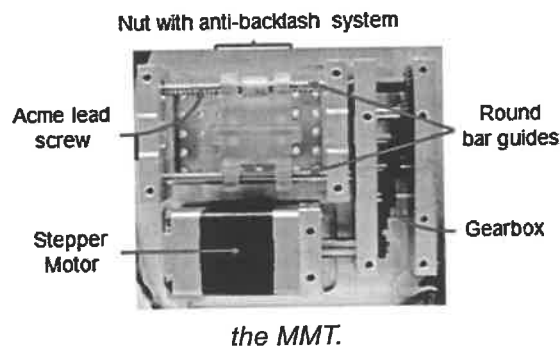


Figure 3. MMT bridge-type prototype.

During the evaluations performed on this prototype, one of the detected errors was related to the fabrication of the lead screw. Thus, a commercial miniature ACME lead screws were implemented. However, it was required to find a solution to conserve the anti-backlash system. Finally, the position of the spindle was changed in order to avoid having it in a cantilever position.

The figure 4 shows the new motion system. In this case, the nut was fabricated by rapid prototyping technology. Helical spur gears were developed using the same technology to improve the characteristics of the gear box.

Figure 4. New motion system implemented on



Moving the spindle from the vertical position to the horizontal position required the increment of the MMT dimensions. The figure 5 shows the final configuration of the MMT. The new characteristics are:

- Dimensions: 110 x 110 x 195 mm³
- Work volume: 18 x 18 x 18 mm³
- Resolution: 93.75 nm per full motor step
- Max. feed rate: 100 mm/min

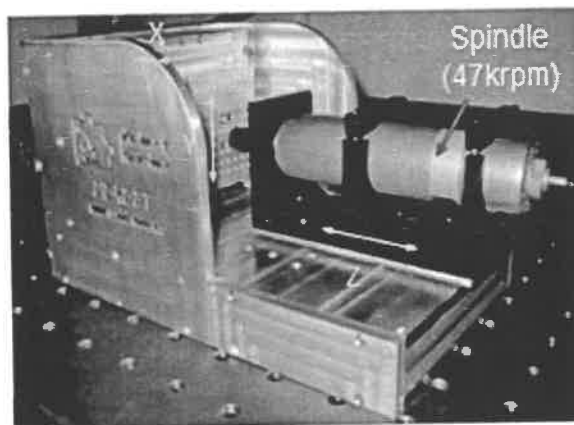


Figure 5. MMT bridge-type redesign.

To evaluate this new system, displacements of 240 microns were realized. The measured average displacement was 239.8 microns, while

the average backlash was under 1 micron. Repeatability is in average of 1.5 microns. This evaluation was performed by a laser interferometer.

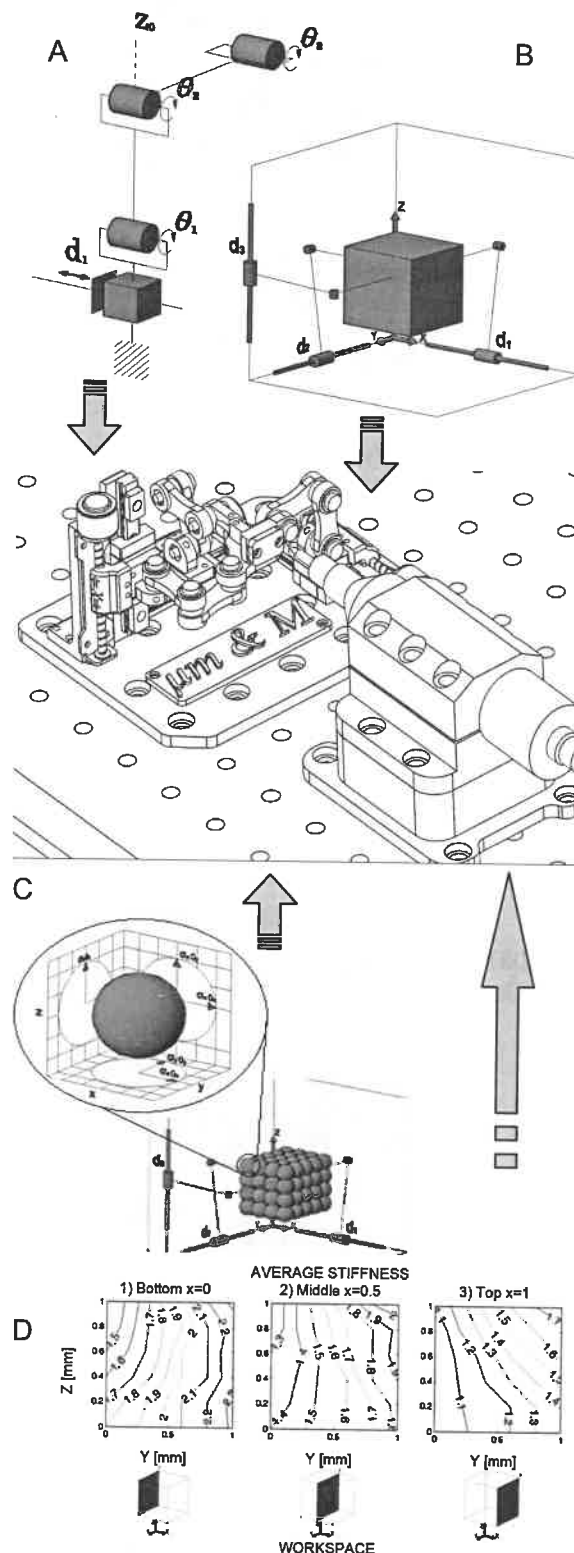
PARALLEL TYPE MICROMACHINE TOOL (2011)

It is well known that, in most of the parallel mechanisms the workspace is not regular, they present a high no linear input-output relation, and the Jacobian matrix, which transforms the joint rates of a mechanism into the end-effector velocity state, is not isotropic. In consequence, the performance (maximum velocity, force transmission, accuracy, stiffness) vary considerably for different points in a Cartesian space and for different directions of a given point. That is, opposite to the requirements in micromachining process, where an isotropic behavior under each direction of load is required, i. e. the ability to transmit homogeneous forces on a regular workspace, and a minimum variation on stiffness values, without a high cost in components and control. Furthermore, in order to perform basic micromachining operations, a pure translational mobility is required.

Although the problems mentioned above, the configuration of micromachine tool based on a parallel mechanism, shown in Figure 6, is symmetric and composed of three identical legs connecting the fixed base to the end-effector. Each leg has a PRRR design, (Figure 6A); with three passive revolute joints (R) and an active prismatic joint (P). Since the orthogonal arrangement of the three linear actuators, provides a regular workspace in parallelepiped form (Figure 6B), then exists a one-to-one correspondence between the input and the output displacements, velocities and forces, and the Jacobian matrix is always isotropic over the entire workspace (Figure 6C). These conditions guarantee an isotropic behavior by each direction of motion and almost constant stiffness values (Figure 6D).

This new prototype, shown in Figure 7, belongs to the first micromachine tool based on parallel configuration developed in Mexico. This prototype has overall sizes of 120 mm x 100 mm x 70 mm and a travel range of $x=15$ mm, $y=15$ mm and $z=15$ mm. It is controlled by a three-axis stepper motor power drives that provide reliable high torque and microstepping. The actuators

are stepping motor with twenty steps per revolution.



Figures 6 A-D. Topology of the micromachine tool based on a parallel configuration.

The theoretical resolution of each axis is 600 nm, within the position error range from 25 μm to 70 μm . Table 1 shows specification of the developed micro machine tool. Figure 8 shows the position error of actuator "X" but, this MMT is under process of characterization.

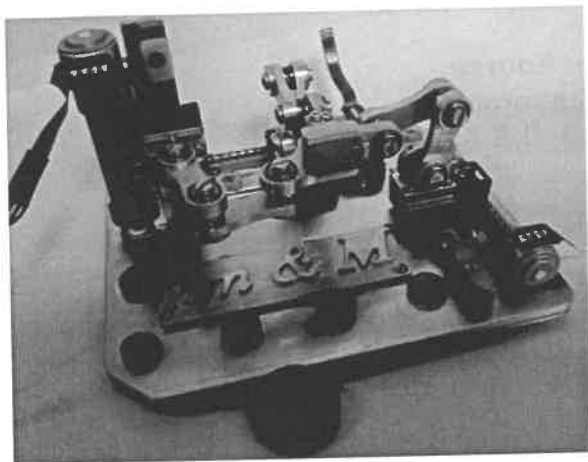


Figure 7. Parallel Type Micromachine Tool near to a 1 mexican peso for scale purposes.

Specification		Description
Stage	Size	120 x 100 x 70 mm ³
	Motor type	20P step motor
	Travel range	15 x 15 x 15 mm ³
	Resolution	0.6 $\mu\text{m}/\text{step}$
	Repeatability	x-axis=3.4 μm , y-axis=3.9 μm , z-axis=4.1 μm
Spindle motor	Speed	180 mm/min
	Max. speed	47000 r/min
	Power	0.15 KW
Cutting tool	Diameter	0.2 mm
	Material	Solid carbide
	Geometry	Square end mill 4 flute

Table 1. The specification of developed MMT based on parallel configuration.

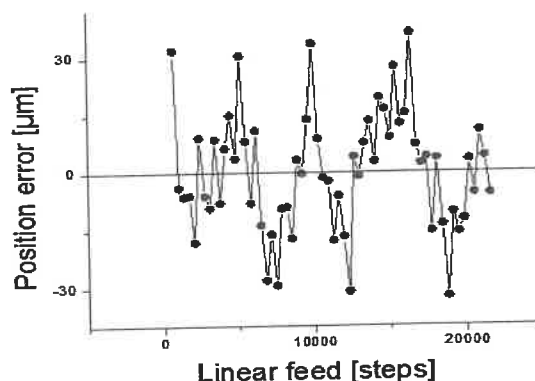


Figure 8. Position error of actuator x.

CONCLUSIONS

This work presents two new prototypes of MMT made in Mexico during the last two years. This prototypes continue under characterization process, and both have shown better results than the previous versions due to the new configurations, the use of commercial components and specific parts designed and manufactured with rapid prototyping methods.

ACKNOWLEDGMENTS

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A NOVEL ULTRAHIGH-RESOLUTION MONOCHROMATOR/ANALYZER SYSTEM FOR AN INELASTIC X-RAY SCATTERING SPECTROMETER WITH 26 PRECISION POSITIONING STAGES

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INTRODUCTION

Many important scientific topics related to the high-frequency dynamics of condensed matter require both a narrower and steeper resolution function and access to a broader dynamic range than are currently available. There are needed for the development of a new inelastic x-ray scattering (IXS) spectrometers with improved energy and momentum resolution.

Three effects in Bragg diffraction of x-rays in backscattering geometry from asymmetrically cut crystals have been observed by Yuri Shvyd'ko et al. in 2006 [1]: (1) Exact Bragg backscattering does not take place at normal incidence to the reflecting atomic planes. (2) A well-collimated beam is transformed after the Bragg reflection into a strongly divergent beam with a reflection angle dependent on the x-ray wavelength (an effect of angular dispersion). (3) Parasitic Bragg reflections accompanying Bragg backreflection are suppressed. These effects provide a radically new method for monochromatization of x rays not limited by the intrinsic width of the Bragg reflection. However, there are many optomechanical design challenges on the road to turning a CDDW optical configuration into a practical instrument [1-5].

Based on the above effects, key components of a novel ultrahigh-resolution IXS spectrometer, monochromator, and analyzer have been designed, built, and tested at the undulator-based beamline 30-ID, at the Advanced Photon Source (APS), Argonne National Laboratory. The new devices are designed to meet challenging mechanical and optical specifications for producing ultrahigh-resolution IXS spectroscopy data for various scientific applications [6,7].

The optomechanical design of the ultrahigh-resolution monochromator and analyzer as well as the preliminary test results of their precision positioning performance are presented in this paper briefly. More details are to be published in a SPIE conference paper [6,7].

THE CDDW X-RAY CRYSTAL OPTICS

As shown in Figure 1, the typical so called CDDW angular-dispersive x-ray crystal optics includes a central ultra-thin CW crystal and a pair of dispersing-elements. The abbreviation CDDW stands for: C – collimating crystal, D – dispersing-element crystal (two D-crystals are used in each CDDW), and W – wavelength-selector crystal.

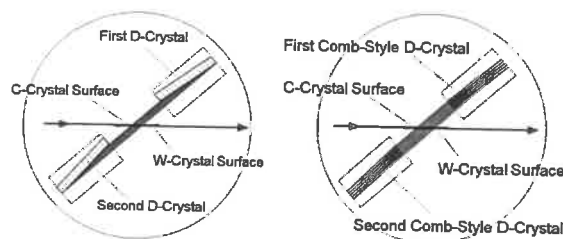


FIGURE 1. Schematic diagram of the typical CDDW x-ray optics for a monochromator with monolithic D-crystals (left) and comb-style D-crystals (right).

The CW crystal performs collimation of the incident x-ray beam and serves as a wavelength-selector. The dispersing elements, which are either strain-free monolithic crystals (Figure 1, left) or comb-style crystals (Figure 1, right) are used to achieve the effect of angular dispersion in exact backscattering from crystal atomic planes. The reflecting crystal planes are almost perpendicular to the working crystal surface in a strongly asymmetric configuration; for instance, with the Si (008) atomic planes for

x-ray energy 9.13 keV. [4-8]. Figure 2 shows a standard three views of a three-dimensional (3-D) model of the precision mechanical structures for a typical CDDW x-ray optics for the ultrahigh-resolution x-ray monochromator with comb-style D-crystals.

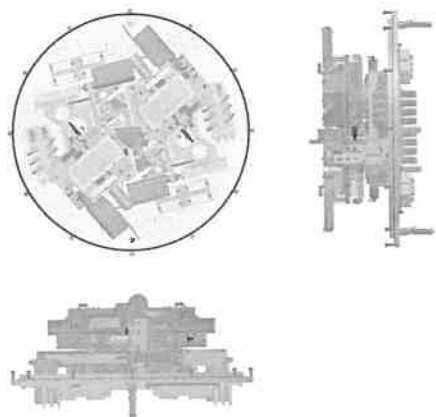


FIGURE 2. A 3-D model of the precision mechanical structures for a typical CDDW x-ray optics for the ultrahigh-resolution x-ray monochromator with comb-style D-crystals.

BASE STAGES FOR THE CDDW X-RAY MONOCHROMATOR AND ANALYZER

The novel IXS spectrometer prototype consists of two similar sets of CDDW x-ray optics: one set for the x-ray monochromator, and the other set acting as an x-ray analyzer. Each of the CDDW optics subassemblies includes 13 positioning stages:

- One base vertical axis linear stage;
- One base horizontal axis linear stage;
- One base horizontal axis rotary stage;
- One base vertical axis tip-tilting stage;
- Two D-crystal primary angular alignment stages;
- Two D-crystal linear alignment stages;
- Two D-crystal tilting alignment stages;
- One CW-crystal tilting alignment stage;
- Two diagnostic detectors in/out stages.

A total of 26 precision positioning stages are integrated in the monochromator/analyzer prototype for the IXS spectrometer instrument. Eighteen of them are APS-designed, customized special stages.

To align the whole CDDW optics to the incoming x-ray beam, the base plate of the CDDW optics is manipulated by a set of base stages, as

shown in Figure 3. The base vertical axis tip-tilting stage is not shown in this figure.

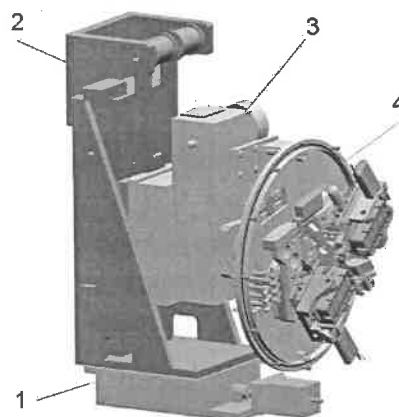


FIGURE 3. A 3-D model of the base stages for the alignment of the whole CDDW optics (except the base vertical axis tip-tilting stage): (1) Base horizontal axis linear stage; (2) base vertical axis linear stage; (3) base horizontal axis rotary stage; (4) base plate of the CDDW optics.

The base rotary stage is a stepping-motor-driven precision goniometer Kohzu™ [9] KTG-15. It is mounted on the carriage of the Kohzu high-load-capacity vertical and horizontal stages KHI-4SK. An APS-designed motorized base vertical tilting stage Z8-7009 provides an interface between the base stages and the optical table, as shown in Figure 4.

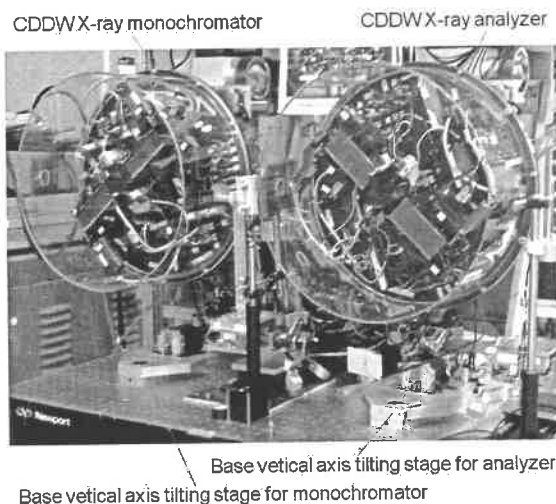


FIGURE 4. A photograph of the prototype of the ultrahigh-resolution IXS spectrometer designed, constructed, and tested at undulator-based beamline 30-ID at the APS.

MODULAR FLEXURE STAGE GROUPS

Two types of flexure elements are applied for the CDDW optics precision alignment and positioning: the commercial flexure pivots and Argonne-developed laminar overconstrained weak-link mechanisms.

Compared with traditional kinematic flexure mechanisms, laminar overconstrained weak-link mechanisms provide much higher structure stiffness and stability. The strain-limited overconstrained mechanism is a stacking structure with thin-metal weak-link sheets manufactured by photochemical machining processes with lithography techniques. The solid complex laminar weak-link structure provides ultrahigh positioning sensitivity and stability [10-12]. Both rotary and linear laminar weak-link elements are commercially available now [13].

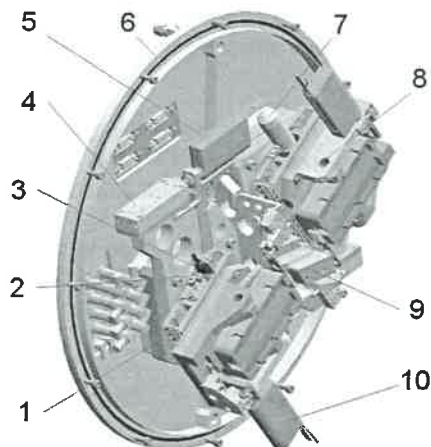


FIGURE 5. A 3-D model of the CDDW x-ray monochromator for the prototype of the ultrahigh-resolution IXS spectrometer constructed at the APS. (1) Linear weak-link modules for Z8-72 stage; (2) rotary weak-link modules for Z8-73 stage; (3) MicroE Systems™ [14] M3500Si grating optical encoder; (4) PI™ [15] closed-loop controlled PZT with capacitance sensor for Z8-73 stage; (5) Newport™ [16] closed-loop controlled Picomotor™ with rotary grating encoder; (6) base plate for CDDW optics; (7) Newport™ PZA-12 piezo motor actuator; (8) commercial flexure pivots; (9) Z8-74 central set of stages for CW crystal; (10) Newport™ closed-loop controlled Picomotor™ with rotary grating encoder.

There are three flexure-based positioning stages stacked together for each D-crystal to control its pitch angle coarse and fine motion, linear fine positioning, and roll angle adjustment as shown in Figure 5. Four sets of such stage modules

have been built for the APS monochromator and analyzer prototypes for the novel IXS spectrometer. To achieve a better than 0.002-arc sec positioning repeatability in an 1.1-degree angular travel range, a compact PZT-driven rotary stage Z8-73 for the D-crystal's pitch angle fine and coarse positioning has been designed using an overconstrained rotary weak-link mechanism developed at Argonne [7-9].

The stage groups for D-crystals are mounted on a vertical base plate with a large angular travel range (up to 180 degrees) around a horizontal rotary axis. A high-stiffness precision linear stage Z8-72, with a 100-nm closed-loop positioning resolution, has been designed with two sets of laminar overconstrained linear weak-link modules perpendicular to each other to provide multidimensional stiffness for the load (up to 2-kg) with various moment directions.

The CW crystal is made ultra-thin (300- μ m) and is mounted on the holder with minimized induced strain to optimize its diffraction effects. Figure 7 shows a 3-D model of the Z8-74 central set of stages for CW crystal alignment.

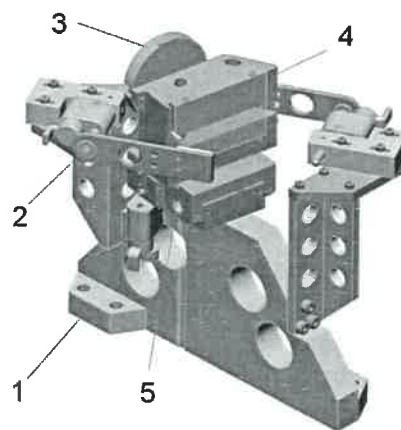


FIGURE 6. A 3-D model of the Z8-74 central set of stages for collimating crystal and wavelength-selector crystal alignment. (1) Base; (2) Picomotor™ stage for diagnostic detector in/out control; (3) US Digital™ [17] X3M multi-axis absolute MEMS inclinometer; (4) thin-film CW crystal holder with roll and pitch angle adjustment; (5) flexure pivots for roll tilting stage.

PRELIMINARY TEST RESULTS AND SUMMARY

A proof-of-principle experiment was performed at the undulator-based beamline 30-ID at the APS with prototypes of the CDDW-optics-based

monochromator and analyzer for an IXS spectrometer in April 2011. Strain-free monolithic D-crystals manufactured at the APS and characterized using x-ray topography [6] were used in this experiment. An energy scan of the CDDW monochromator around 9.13-keV was performed by a simultaneous change of the angles of the D-crystals. A 0.65-meV combined energy resolution of the monochromator-analyzer pair was demonstrated in the in-line configuration, as well as sharp tails of the spectral distribution function [5]. The tails were very steep, more than 100 times steeper than the tails of a Lorentzian distribution with the same FWHM.

The design and construction of a complete prototype of the IXS spectrometer with sample stages, goniometers, and collimating optics, etc., as well as the development of ultrahigh-quality comb-style D-crystals is in progress at the APS. The technical details of the CDDW x-ray optics and further development of the CDDW-optics-based IXS spectrometer will be published in separate papers.

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- [16] Newport is a trademark of the Newport Corporation, Irvine, CA, USA.
- [17] US Digital is a trademark of the US Digital, Inc., Vancouver, WA, USA

MODELING AND MEASUREMENT OF H-BOT KINEMATIC SYSTEMS

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INTRODUCTION

Due to the ever increasing demand for high productivity which implies high dynamics at reduced costs additional efforts are required in the search for alternative structural designs. This leads to the permanent investigation of alternative machine concepts for the application in fields, where standard, serial concepts have been used so far. Even if alternative kinematics are already used as is the case of 3D pick and place applications, other parallel structures have to be studied in order to improve the system's overall capabilities. In this paper a parallel kinematic structure called H-Bot has been investigated [1] and realized comprising modeling and measurement. The results of both areas of investigation are shown and discussed.

THE H-BOT CONCEPT

This H-Bot concept represents the interaction of two rotary drives which are connected by a single H-shaped circumferential timing belt around two staggered linear axes in a gantry-type like configuration, see figure 1. The kinematics of the H-Bot mechanism formulated in (1), where x and y are the TCP- and q_1 and q_2 are the drive-coordinates in radians and r_d is the drive's pulleys radius:

$$\begin{Bmatrix} x \\ y \end{Bmatrix} = r_d \begin{bmatrix} -1 & 1 \\ -1 & -1 \end{bmatrix} \begin{Bmatrix} q_1 \\ q_2 \end{Bmatrix} \quad (1)$$

Due to the fact that the drives do not have to be moved the achievable dynamic values can be quite high. Another aspect of this concept in contrast to delta-robot like kinematics is that the weight of the work load is carried by the linear guideways and not by the drives. Also, the working envelope, the required transformations and the uniformity of the distribution of properties are far easier to handle for industrial

use than the often odd shaped working envelopes, complex kinematics and the strongly nonlinear property distribution of alternative concepts.

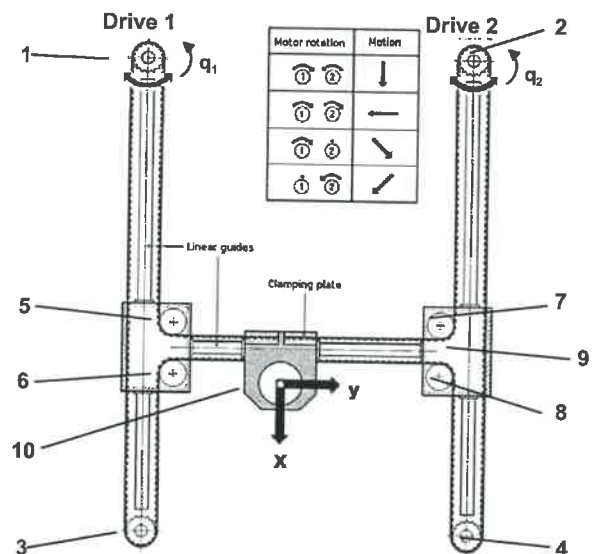


FIGURE 1. Schematic of the H-Bot configuration and the kinematic [3], numbering of bodies.

MEASUREMENT RESULTS

On a first prototype a series of measurements covering geometric properties could be carried-out. Figure 2 shows exemplarily the results of a circle test. As opposed to measurements on serial axis concepts the systematic deviations show additional systematic effects: At axis reversal similar effects are noticed as are typical for axis having semi-closed position feed-back. In addition to these at the axes reversals similar affects are visible at positions on the $\pm 45^\circ$ diagonals. These effects are caused by drive reversal, where stiction occurs on the driven pulley and/or motor bearing in combination with backlash of the belt.

Based on these first measurements on the prototype, a number of improvements could be achieved using feedforward torque computation. By this effects of friction and inertia could strongly be reduced as can be seen in Figure 3. Especially the effects on the ± 45 locations could be reduced significantly.

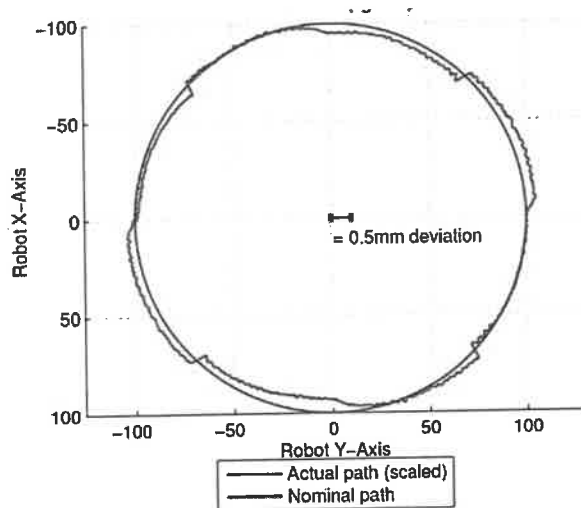


FIGURE 2. Result of a circle test in CCW on the H-Bot prototype without application of compensations, radius 100mm

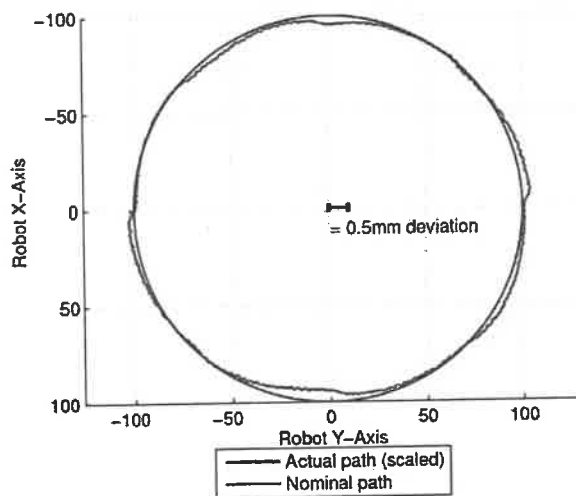


FIGURE 3. Result of a circle test in CCW on the H-Bot prototype with controller compensations, radius 100mm

For pick and place applications that the robot is designed for, position accuracy is only important at the end of a motion. In Figure 4 and 5 the results of a positioning measurement according to ISO 230-2 are shown. As can be seen, a quite good repeatability is obtained, when regarding

unidirectional approach of measuring positions. The reversals are quite large and slightly position dependent. The position deviations show a systematic slope, which could be reduced, when applying linear pitch error compensation.

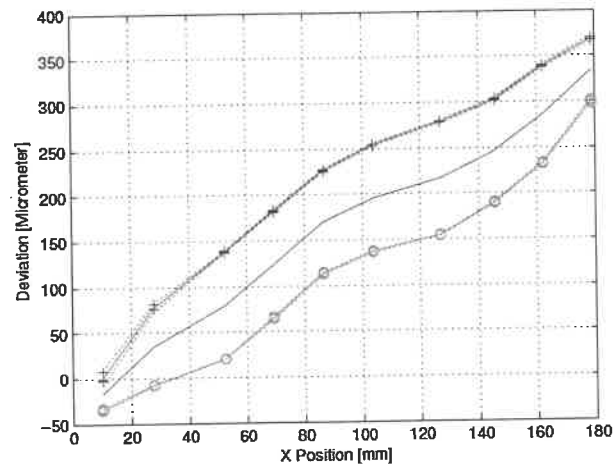


FIGURE 4. Positioning results in X-direction on the H-Bot prototype according to ISO 230-2.

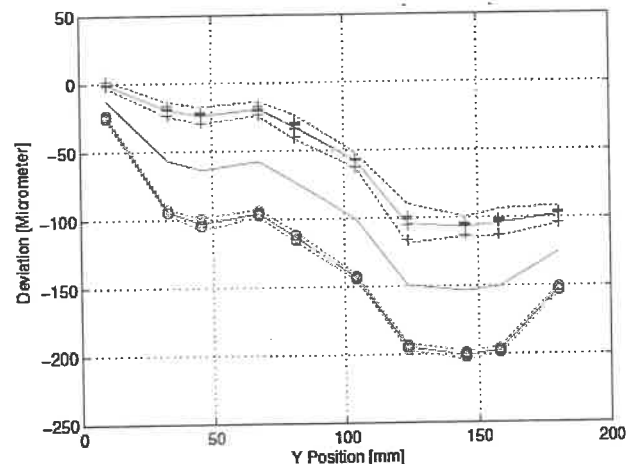


FIGURE 5. Positioning results in Y-direction on the H-Bot prototype according to ISO 230-2.

In Figure 6 2D-measurement results for a series of pendulum motions in Y-direction over 100 mm 0.4 m/sec programmed feed rate, dwell times at 0 / 100 mm: 2 seconds.

Also in this case a high repeatability between the different runs is given. As for the positioning measurements a large, in this case lateral hysteresis can be seen. Friction in combination with limited stiffness of the belt can be seen as reason for this behavior.

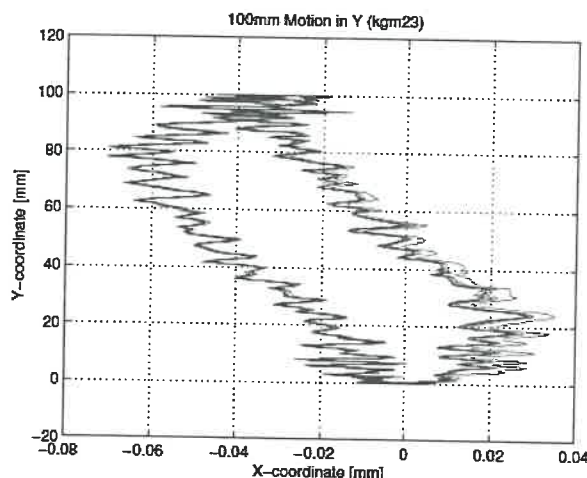


FIGURE 6. 2D-Path measurements (cross-grid) for a series of pendulum movements over 100mm in Y-direction at 0.4m/sec feed rate.

MODELING

In order to gain knowledge about the systems behavior before a prototype has been built and measurements can be made, appropriate modeling can be a very useful way to eliminate uncertainties concerning technical decisions [2]. In a first step, the kinematic parameters can be used to elaborate the relationship between the movements of the drives, the other components of the mechanism and the TCP also known as Jacobian matrix of the system. Taking into account the inertia values of the components and the friction parameters, the major requirements for the drives needed for dimensioning are obtained.

In a second step, the system shall be described for analyses regarding static loads and behavior in the frequency-domain. Here, additionally to the previous step, stiffness and damping values are required for the state-space representation of the system. At this stage, the linear control of the drives can be added.

In a third step, the system is finally completed with the non-linearities such as coulomb friction, backlash, and quantization of measurement systems e.g.

Regarding the H-Bot configuration shown in FIGURE 1, there are ten bodies which primarily define the systems behavior. For the

investigation of the planar X-Y-behavior only 22 degrees of freedom have been selected:

- C-DOFs of all bodies,
- X- and Y-DOFs of bodies 5 to 10.

In this way, the yaw movements of the X-cross beam carrying the y-slider are included in the model.

The pulleys are connected by linear stiffnesses representing the belt. Linear stiffnesses in Y- and X-direction connect the X-cross beam with the inertial system and the Y-slider with the X-cross beam respectively.

Based on the inertia, stiffness and damping values and the configuration of the components a state space representation has been elaborated.

As inputs the torque of the two drives, nonlinear friction and external loads acting at the TCP are available.

As outputs of the state-space system the TCP-movements and the signals for the semi-closed or alternatively closed-loop position feed back are used for the simulation.

Alternative Model

In Figure 7 an alternative way to model an H-Bot like structure as robot model [4] is presented:

Here, the yaw movements of the X- and Y-slider and the single pulleys have been omitted.

In this case the focus was laid on the application of state space control using acceleration values obtained at the TCP [2].

Special attention was given on the backlash of the belts which are responsible for the effects at the $\pm 45^\circ$ locations noticeable in Figure 2.

Regarding the effort required for the optimisation of the compensator parameters to obtain the promising results shown in Figures 3 to 5, the availability of those models shall represent a significant support. Also the further improvement/ development of the system comprising modifications of configuration parameters such as dimensional proportion parameters (lengths / widths of sliders e.g.) or use of alternative feed-back system parameters are significantly simplified by the availability of these models.

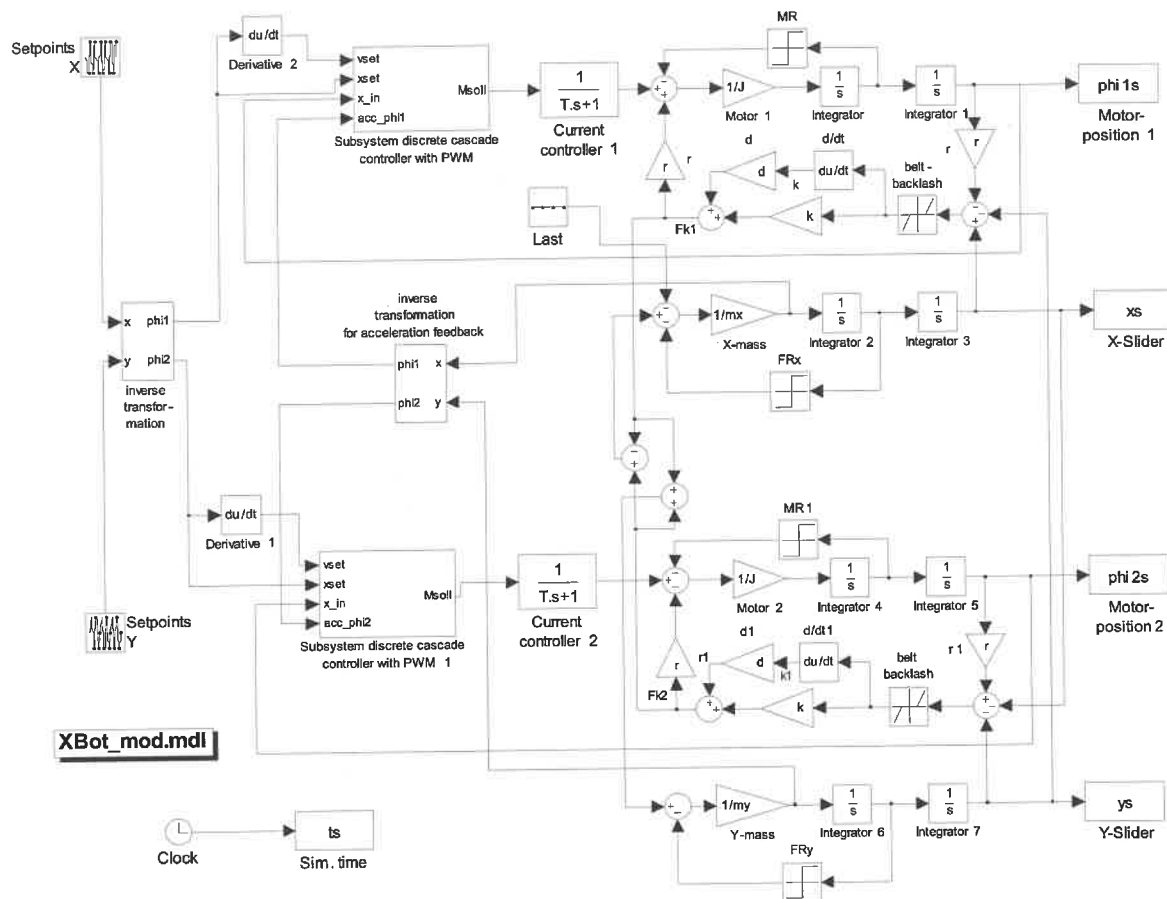


FIGURE 7. Mechatronic Simulink model of a X-Bot kinematic (similar to H-Bot) containing drives and sliders

CONCLUSION

The principal systematic effects of a new kinematic concept have been investigated using measurements and simulations.

The H-Bot prototype shows very good repeatability and systematic effects due to friction and elasticity.

On the simulation side different modeling approaches have been discussed. The models derived here will significantly support the further development of H-Bot like configurations.

ACKNOWLEDGEMENT

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DESIGN, FABRICATION, AND TESTING OF A MOVING MAGNET ACTUATOR FOR LARGE RANGE NANOPositionING

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ABSTRACT

Moving magnet actuators are promising candidates for achieving nanometric motion quality over a large (10mm) motion range. Large range nanopositioning capability addresses an important instrumentation need in several areas of nanotechnology. In pursuit of this goal, we present an optimal moving magnet actuator design integrated with a double parallelogram flexure bearing and thermal management system. The proposed design is systematically optimized in the context of fundamental actuation- and system-level performance trade-offs that we have identified. The overall system is custom-fabricated and tested to validate the desired actuation performance. Preliminary closed-loop results highlight the proposed actuator's potential for large range nanopositioning.

INTRODUCTION AND MOTIVATION

Large range (~ 10 mm) nanopositioning systems are becoming increasingly desirable in applications such as scanning probe microscopy, nanometrology, nanolithography, hard-drive and semiconductor inspection, and memory storage [1-2]. However, most existing nanopositioning systems are generally limited to a motion range of a few hundred microns [2-3], in part due to actuation challenges. The voice coil actuator (VCA), which is a single-phase electromagnetic actuator, has stood out among other options such as piezoelectric, inchworm-type, and multi-phase electromagnetic actuators, for its non-contact, frictionless and cog-free motion characteristics and sufficient motion range suitable for large range nanopositioning [4-5]. However, in a VCA, the non-deterministic disturbance due to the moving connecting wires and heat dissipation from the coil connected to the motion stage degrade the overall motion quality (resolution, precision and accuracy).

While the operation of a moving magnet actuator (MMA) is similar to that of a VCA, its construction is different in that it employs non-moving coils and a moving magnet. FIGURE 1

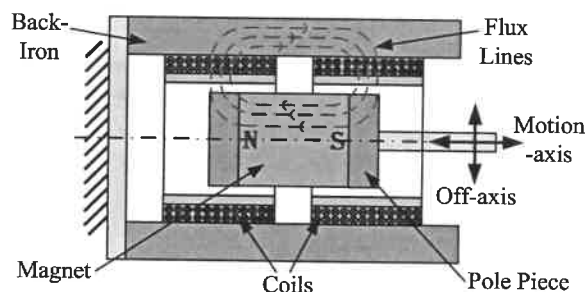


FIGURE 1. Moving Magnet Actuator Schematic.

illustrates a traditional MMA configuration, featuring an axially-oriented cylindrical magnet and iron pole pieces that are connected to the motion stage of the bearing, which serve as the "mover". The "stator" consists of a tubular back-iron, along with two oppositely wound coils that are connected in series. This configuration gives MMAs two distinct advantages over the VCA: I. The non-deterministic disturbance due to the moving cables is eliminated, making MMAs truly non-contact actuators. II. Since the coils are connected to the static back-iron as opposed to the mover, the heat generated due to resistive losses in the coils remains physically separated from the motion stage. Given these attributes, MMAs are promising candidates for large range nanopositioning.

The objective of this research is to systematically develop and demonstrate an optimal MMA design that takes into account I. Physical performance trade-offs, as described further below, and II. MMA-specific challenges including greater non-uniformity in force over stroke and off-axis instability of the mover [6].

PERFORMANCE TRADE-OFFS

An MMA, double parallelogram flexure bearing, and thermal management system were concurrently designed to meet the following system-level performance criteria: I. Maximize the first natural frequency (or open-loop bandwidth) since this is closely related to the achievable speed and disturbance rejection of the motion system in closed-loop operation. II. Minimize the power consumption of the actuator

and effectively dissipate the generated heat, which is detrimental to the motion system's performance. **III.** Maximize the uniformity of force with respect to position over the motion range (i.e., actuator stroke), since non-uniformities can compromise the closed-loop tracking performance of the motion system. **IV.** Ensure that the bearing counteracts any off-axis attraction between the mover and the back-iron to prevent the mover from pulling in sideways to the stator.

Simultaneously satisfying all of these criteria without any physical limits is not possible, however; for example, the force output of an MMA can be increased by either increasing the moving mass or by increasing the quasi-static power limit, both of which are undesirable. To quantitatively determine these trade-offs, a lumped-parameter model has been developed [7], which shows that for the traditional actuator configuration of FIGURE 1, the actuation force (F) remains directly proportional to the square root of the actuator moving mass (m_a) and the square root of power consumed (P), irrespective of the geometric size of the actuator. This may be expressed as:

$$\frac{F}{\sqrt{P}\sqrt{m_a}} = \frac{K_t}{\sqrt{R}\sqrt{m_a}} = \beta \text{ (constant),}$$

where K_t is the force constant and R is the coil resistance. When employed in a flexure bearing-based motion system, an important consequence of the above relation is that it places a limit on the overall open-loop bandwidth (ω_n), given a desired motion range (Δ), power consumption limit (P_{max}), and mass of the motion stage (m):

$$\omega_n^2 < \beta \frac{\sqrt{P_{max}}}{\Delta} \frac{\sqrt{m_a}}{m + m_a}$$

This, in turn, limits the achievable bandwidth and disturbance rejection (and therefore positioning resolution) in a closed-loop setup [8].

MMA DESIGN AND FABRICATION

As shown above, it is evident that in order to maximize the open-loop bandwidth of the motion system based on an MMA and flexure bearing, one has to maximize the constant β while keeping the moving mass as small as possible. In this work, the motion range was set to be

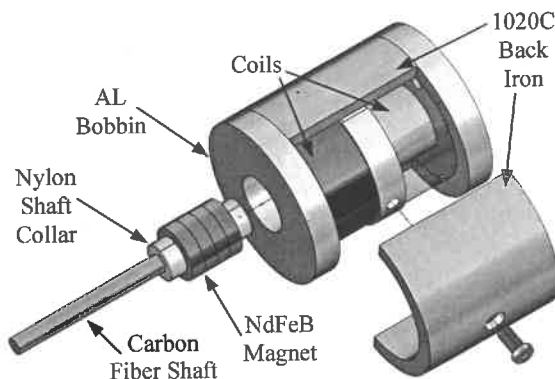


FIGURE 2. Exploded View of the MMA.

$\pm 5\text{mm}$, with a power and voltage limit of 20W and 25V, respectively, based on a custom-made low noise drive amplifier and the capability of the thermal management system (described later).

The geometric design of the MMA (i.e., dimensions of the magnet, back-iron, and coil) was then carried out in a systematic manner to maximize β and minimize the moving mass [7]. This resulted in the following actuator specifications: force constant of 30N/A, actuator moving mass of 106g, coil resistance of 43 Ω , force-stroke non-uniformity less than 9%, and a β value of 14 $\sqrt{\text{Hz}}$.

Based on this design, a prototype of the MMA was fabricated and assembled. The components of the prototype are shown in an exploded view in FIGURE 2. A Neodymium-Iron-Boron (Grade N52) axial magnet was selected for its high remanent magnetization (1.45T), which provides a high force constant. The magnet is maintained below its Curie temperature (80°C) via a thermal management system, as described later. To minimize the moving mass, the magnet is mounted on a hollow carbon fiber shaft using Nylon sleeve collars. The coil was wound using 25AWG copper wire on an Aluminum bobbin. AL6061 was chosen for its good machinability and high thermal conductivity, and it also acts as a shorted turn, which reduces the coil inductance. The back-iron, turned from 1020C steel, exhibits a high saturation flux density of 1.6T. Additionally, it was designed as two symmetric halves so that the rest of the actuator can first be easily assembled without being affected by the strong permanent magnet field.

THERMAL MANAGEMENT SYSTEM

To make sure that the heat generated by the MMA coils will not adversely affect the

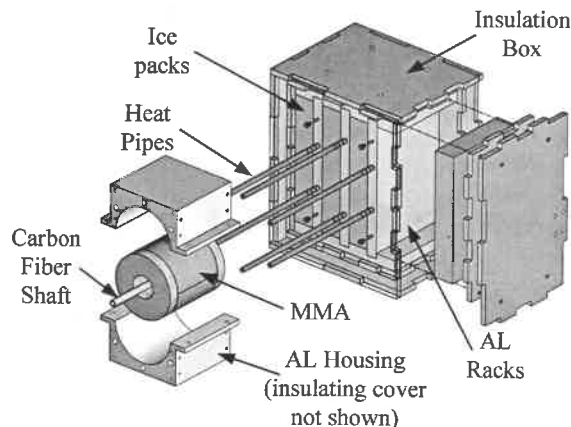


FIGURE 3. Thermal Management System.

remanence of the permanent magnet, cause distortions in the flexure bearing, or compromise the accuracy of the position sensor, a passive thermal management system (TMS) was designed and incorporated with the actuator, as shown in FIGURE 3. It provides an effective way to transfer heat from the MMA coils to an ice sink via heat pipes. This method is advantageous as compared to other convective heat dissipation methods, which may lead to air flow-induced vibrations.

A lumped-parameter thermal network model was developed to evaluate the steady-state temperature of the coil bobbin. Using this, the critical components of the TMS (heat pipes, ice packs, and Aluminum racks) were designed in order to ensure that the steady-state coil bobbin temperature remains near room temperature for at least 4 hours of operation under constant 20W power input to the actuator. The experimental results of the TMS under these conditions are shown in FIGURE 4.

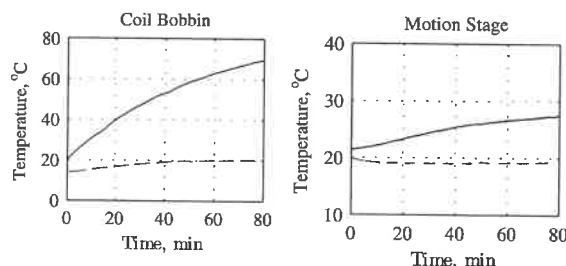


FIGURE 4. Temperature Rise with (---) and without (—) Thermal Management System.

BEARING DESIGN AND FABRICATION

A single-axis symmetric double-parallelgram flexure bearing (FIGURE 5) was selected to provide motion guidance for the MMA for its

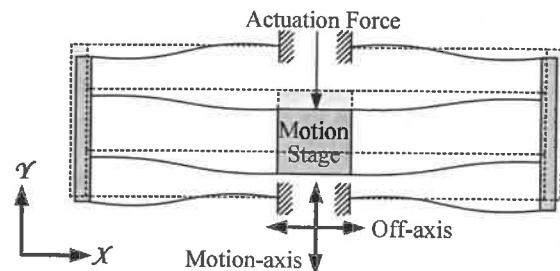


FIGURE 5. Symmetric Double Parallelgram Flexure Bearing.

linear motion direction stiffness, relatively high off-axis stiffness, and frictionless and backlash-free motion. The translational stiffness of the flexure bearing in the motion direction (K_y), in-plane off-axis direction (K_x) and out-of-plane off-axis direction (K_z) are given by:

$$K_y \approx \frac{WT^3E}{L^3}; K_x \approx \frac{WTE}{L}; K_z \approx \frac{W^3TE}{L^3}$$

where E is the Young's modulus of the material, and W , T and L are the width, thickness and length of the beam, respectively.

The minimum beam thickness and maximum beam depth were limited by the capability of the water-jet machining process and were set to be 0.75mm and 25.4mm, respectively. The beam length was chosen to be 80mm to provide a ± 5 mm motion range with a motion direction stiffness of 3.43N/mm. With AL6061 as the bearing material, the factor of safety against yield was estimated to be 3.96.

The negative (destabilizing) stiffness of the off-axis force between the magnet and the back-iron was calculated via FEA to be 1.3N/mm near the nominal equilibrium position, and the positive (stabilizing) stiffness of the bearing in the X- and Z-directions was found to be 149.6N/mm and 70.6N/mm, respectively, thereby ensuring the off-axis stability of the MMA.

EXPERIMENTAL SETUP

The single-axis motion system prototype, comprising the MMA, flexure bearing, and thermal management system, is shown in FIGURE 6. A custom-made current driver was built to provide direct control of the actuation force. An off-the-shelf linear optical encoder (5nm resolution) was used for position measurement and feedback. The system was

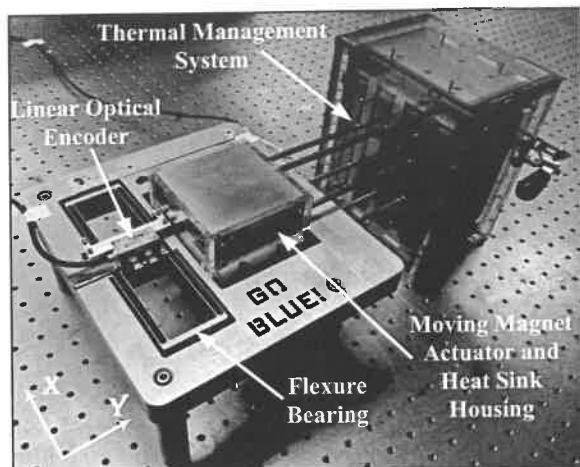


FIGURE 6. Nanopositioning System Prototype.

tested to demonstrate a 36Hz open-loop bandwidth as shown in FIGURE 7.

POSITIONING PERFORMANCE

The motion quality of a nanopositioning system is ultimately dependent on the design and performance of a closed-loop controller. In a preliminary effort, this motion system was tested for its point-to-point positioning performance with a lead-lag controller. The steady-state positioning noise was found to be less than 4nm (RMS) over the entire 10mm range of motion (FIGURE 8). In this experiment, the motion stage is actuated in steps of 2.5mm and 20nm (*inset*). These preliminary results show promise for the MMA to be used for large range nanopositioning systems, as it provides bandwidth comparable to VCAs, however with better thermal management and truly non-contact operation, leading to superior positioning performance. Ongoing work includes: I.

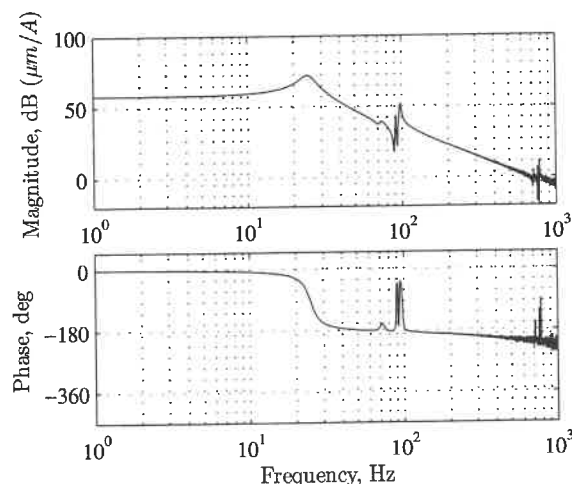


FIGURE 7. Open-Loop Frequency Response.

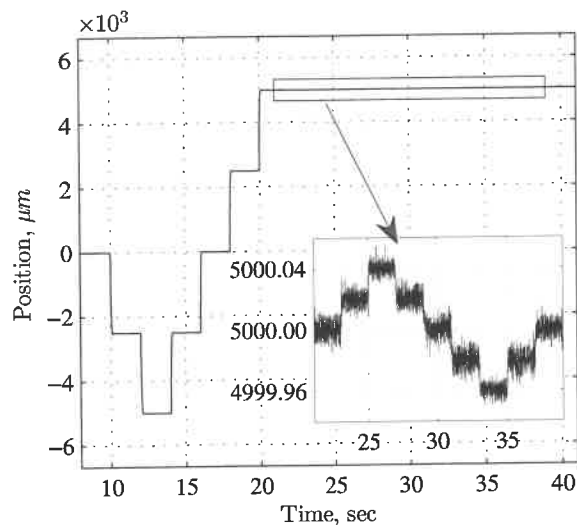


FIGURE 8. Motion Stage Position Response.

Derivation of accurate closed-form models of the MMA's magnetic circuit. II. Design of novel MMA configurations with higher values of β to further improve system bandwidth and motion quality.

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Kinematic analysis and control of 3-DOF parallel manipulator.

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1. Introduction

Recently, Advances in nano science and technology need precision positioning. It is should be moved with sub-micron resolution. Accordingly, many types of manipulator and stage have been developed. Generally, the stage moves on a plane. In contrast, the manipulator moves is s space. The parallel manipulators have received attention because of their properties of high accuracy, large stiffness, fast speed, compactness, and high payload capability. Many parallel manipulators with specified number and type of DOF (Degree Of Freedom) have been proposed. Many researchers have made an effort to implement 6-DOF parallel manipulators. However, they suffer from the problems of complex forward kinematics, relatively small useful workspace, and complicated multi-DOF joints. To overcome these shortcomings, parallel manipulators with 3 DOFs have been investigated. By connecting the two 3-DOF manipulators, they configured the 6-DOFs. The parallel manipulator with three pure rotational DOFs were presented by Kong and Gosselin [1], Di Gregorio [2], and Hess-Coelho [3]. Kim and Tsai [4] designed a Cartesian parallel manipulator that employs only revolute and prismatic joint. In this paper, 3-UPU (Universal-Prismatic-Universal) manipulator [5] will be presented.

2. Geometric modeling

The manipulator consists of an equilateral base plate, a movable equilateral platform, and three

legs that connect the aforementioned two plates with each other.

In this 3-DOF parallel mechanism, the kinematic chains associated with the three identical legs consist from base to platform of a fixed universal joint, an actuated prismatic joint and universal joint attached to the platform.

3. Inverse kinematics

The positions of the universal joints on the base plate of the manipulator are denoted as B_1^o , B_2^o and B_3^o . They are with respect to the reference frame (O) at the stationary platform.

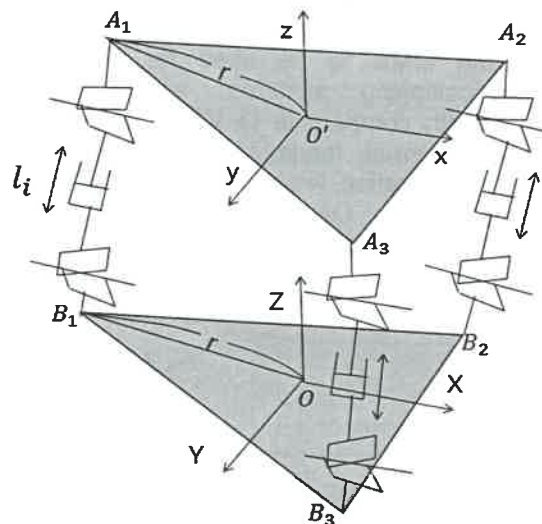


FIGURE1. Geometric configuration of 3-UPU manipulator

Similarly, the positions of the others universal joint on the moving platform of the manipulator are denoted as $A_1^{o'}$, $A_2^{o'}$ and $A_3^{o'}$ with respect to the reference frame at the upper platform (O'). Using the roll (ϕ) and pitch (θ) angles, the angular transformation matrix of the upper platform with respect to base can be expressed as

$$R = \begin{bmatrix} c\theta & s\theta s\phi & s\theta c\phi \\ 0 & c\phi & -s\phi \\ -s\theta & c\theta s\phi & c\theta c\phi \end{bmatrix} \quad (1)$$

Where, $c\theta = \cos\theta$, $s\theta = \sin\theta$, $c\phi = \cos\phi$ and $s\phi = \sin\phi$

Here, and after, s and c are used for the short hand notations of sine and cosine functions respectively. The length of the three legs of the manipulator can be found by using the previous definitions of the joint positions and the geometry as described below:

$$A_i^{o'} = O'^o + R \cdot A_i^{o'} = B_i^o + l_i \quad (2)$$

$$l_i = O'^o + R \cdot A_i^{o'} - B_i^o \quad (3)$$

Here $A_i^{o'}$ denoted the position of the i^{th} universal joint on the upper platform with respect to the center of the base and O'^o denoted the position of the upper platform center with respect the base center. It should be noted that, due to the constraints on the manipulator, the upper platform can only make Z-tip motion with respect to the base plate. The vector joining the i^{th} universal joint on the base to the i^{th} universal joint on the upper platform is defined by l_i with respect to the reference frame at base. Thus,

$$\begin{aligned} l_1 &= \sqrt{C_{11}^2 + C_{12}^2 + C_{13}^2} \\ l_2 &= \sqrt{C_{21}^2 + C_{22}^2 + C_{23}^2} \\ l_3 &= \sqrt{C_{31}^2 + C_{32}^2 + C_{33}^2} \end{aligned} \quad (4)$$

Where, C_{ij} ($i=1, 2, 3$ and $j=1, 2, 3$) are

$$C_{11} = [1 - c\theta + \sqrt{3}s\theta s\phi] \frac{r}{2}$$

$$C_{12} = [c\phi - 1] \frac{\sqrt{3}}{2} r$$

$$C_{13} = [s\theta + \sqrt{3}c\theta s\phi] \frac{r}{2} + Z_0$$

$$C_{21} = [1 - c\theta - \sqrt{3}s\theta s\phi] \frac{r}{2}$$

$$C_{22} = [1 - c\phi] \frac{\sqrt{3}}{2} r$$

$$C_{13} = [s\theta - \sqrt{3}c\theta s\phi] \frac{r}{2} + Z_0$$

$$C_{31} = [c\theta - 1]r, C_{32} = 0, C_{33} = Z_0 - rs\theta$$

Here, r is the distance from center to vertex of each triangle and Z_0 is the initial height of the platform.

4. Fabrication of 3-UPU manipulator

In order let the manipulator have large actuating force and high precision, the mechanism adopts three PZTs (Piezoelectric actuators). Table 1 shows the specifications of each PZT.

Table 1. Specifications of PZT

Displacement	45 μm
Resolution	0.9 nm
Blocked force	3,000 N
Stiffness	75 N/ μm
Resonant frequency	9 kHz

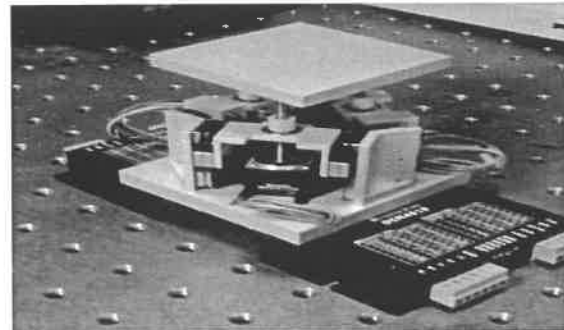


FIGURE 2. Fabricated 3-UPU manipulator

5. Experiments

Based on the kinematics and mechanism design, 3-UPU manipulator was controlled as shown in Fig. 2. The system schematic is shown in Fig. 3.

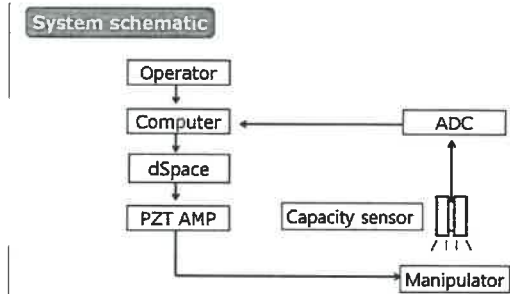


FIGURE 3. System schematic

The position sensor is a capacitive sensor(ADE Tech) which has 1.18 nm resolution and 25 μm range.

First time, the experiment is conducted as presented in Fig. 4 that capacitive sensor probe is under the driving platform. As a result, resolution was in Fig. 5 and Table 2. The Z-direction resolution 200 nm and rotational resolution for X and Y axes was 10 μrad and 8.3 μrad , respectively.

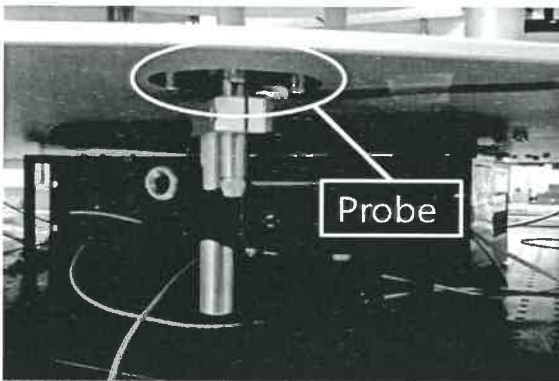


FIGURE 4. Experimental setup for resolution test.

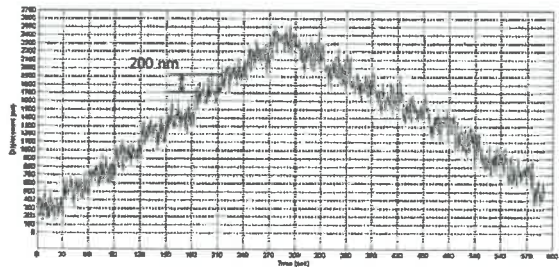


FIGURE 5. Z-directional resolution.

Table 2. Resolution of the 3-UPU manipulator

T_z	200 nm
θ	10 μrad
ϕ	8.3 μrad

Due to the unsatisfied resolution, previous design and measurement method is modified. To prevent deflection for base, thickness is increased. Also, when capacitive sensor measures, it is also exposure to parasitic vibration. In order to reduce the vibration, capacitive sensor is set up on the base using a rigid jig as in Fig. 6

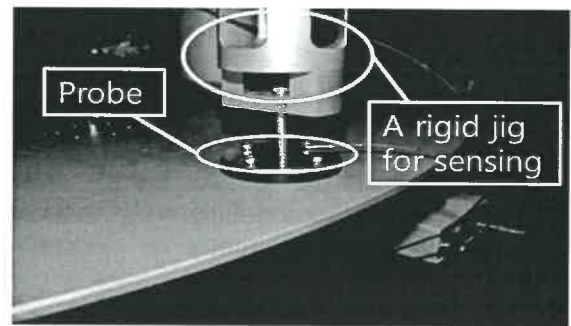


FIGURE 6. Modified experimental setup for resolution test

As a result, resolution improve as shown in Fig 7 and Table 3. The Z-directional resolution is 100 nm and rotational resolution for X and Y axes was 7 μrad and 6.8 μrad , respectively.

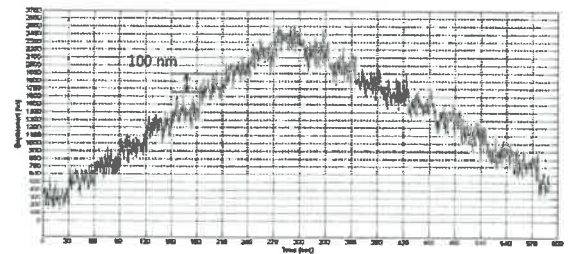


FIGURE 7. Z-directional resolution after modification

Table 3. Resolution of the 3-UPU manipulator (Modify)

T_z	100 nm
θ	7 μrad
ϕ	6.8 μrad

6. Control

PZT has advantage in large stiffness, fast response, unlimited resolution and immunity from magnetic noise. In other hand, PZT has disadvantage in some nonlinearities (creep) and hysteresis. Hysteresis has adverse effect on precision manipulator. Close-loop control is able to remove the hysteresis. To improve positioning precision PID control is introduced as shown in Fig. 8

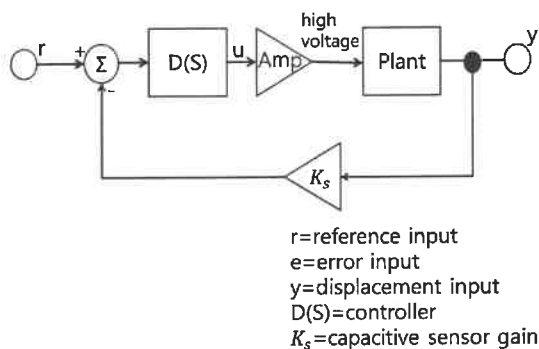


FIGURE 8. Block diagram for close-loop PID control.

After, adequate gain tuning, the hysteresis is eliminated in Fig 8.

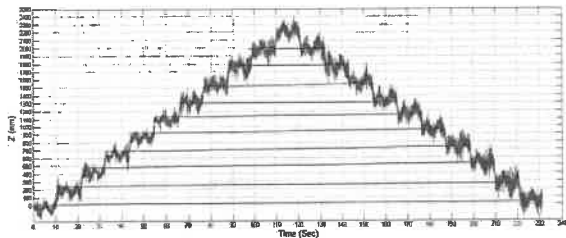


FIGURE 8. Experiment result for reduced Hysteresis.

7. Conclusion

In this paper 3-DOF parallel manipulator was developed. To control the mechanism, kinematic analyses are performed. Base on the design process, 3-UPU manipulator is fabricated and tested. The first experimental are unsatisfiable resolution as shown in Table. 2. However, after modifying previous design, resolution is satisfied as shown in Table. 3. Also, to remove the hysteresis of PZT a close-loop PID control is applied.

The different value between the demand and experiment are caused by error in mechanical works, assemble, nonlinear properties of materials. If the vibration is decreased by improving stiffness and the error from the inaccurate assembling is minimized, the 3-UPU manipulator will have advanced resolution and stability.

8. ACKNOWLEDGEMENT

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ANALYSIS OF THE COSINE ERROR IN MEASUREMENTS USING THE NANOPositionING AND NANOMEASURING MACHINE

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INTRODUCTION

The design of nanocoordinate measuring machines has been predetermined by the increasing demand for the accurate measurement of meso-scale objects with nanometer uncertainties. These precise devices are characterized by nanometer-precise positioning and measurement. A specific example of these devices is the Nanopositioning and Nanomeasuring Machine (NPMM) developed at the Institute of Process Measurement and Sensor Technology (IPMS) in cooperation with SIOS Messtechnik GmbH. The NPMM has a measuring range of $(25 \times 25 \times 5) \text{ mm}^3$. Its guaranteed full-range resolution is less than 0.1 nm with a repeatability of less than 0.3 nm.

The combined positioning uncertainty for a 3-D measurement task is approx. 7 nm, which stems from several factors. The integrated metrological concept yields Abbe-offset-free measurement values in all three axes directly traceable to national and international standards because of the use of stabilized fiber-coupled He-Ne laser interferometers as the measuring devices. The arrangement of these interferometers carries out an intersection of the measuring beams in one virtual point. Furthermore, angle sensors based on the autocollimation-principle in the x- and y-axis help to control guiding errors during the measurement. In addition, there is also a reduction in the number of degrees of freedom in comparison with commercial coordinate measuring machines (CMM). The advantage of the NPMM is that the sample is moved and the probe system simply acts as a zero-point indicator [1], [2]. This so-called sample scanning mode and the fact that the NPMM uses an arrangement of decoupled measuring devices leads to 6 (+3) degrees of freedom (6 in each measuring axis, 3 in the orthogonality of the coordinate system). The schematic design of the metrological concept is shown in the FIGURE 1. Nevertheless, in addition to the overall well-designed mechanical setup, the metrological performance is proven rather by the measurement uncertainty than by the best estimate. It is the

responsibility of a user to determine an effective uncertainty budget, which ideally includes all the influences to the measurement task and provides the final measurement result. By establishing the uncertainty budget in accordance with the *Guide to the Expression of Uncertainty in Measurement* (GUM) [3], combining all error sources could be a complex process because each measurement task depends on specific environmental influences, systematic effects and measurement strategies. It is a common practice to use approaches in agreement with national and international metrological institutes to provide the measurement result and the associated uncertainty [4], [5]. This contribution deals with another strategy which implies using a vectorial modular model (VMM) to express the uncertainty budget to evaluate the relevant metrological quantities. In consensus with the GUM the uncertainty analysis is conducted with the use of a Monte Carlo simulation. Because of the above-noted complexity of combining all error sources the influence of the cosine error on the measurement result is to be described as an example.

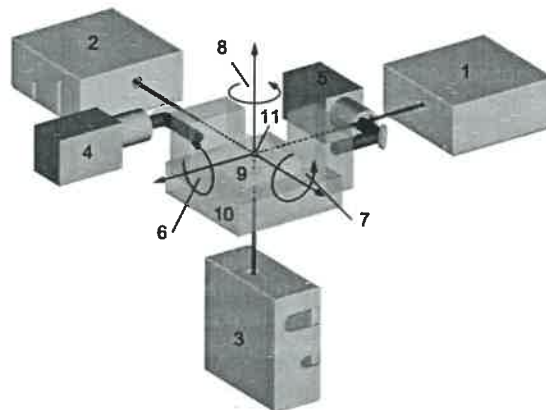


FIGURE 1. Schematic design of the NPMM with decoupled measuring devices (1, 2, 3 – x-, y-, z-interferometer; 4, 5 – angle sensors; 6, 7, 8 – roll-, pitch-, yaw-angle; 9 – probe; 10 – corner mirror; 11- virtual intersection point).

METROLOGICAL VECTORIAL MODELING

As mentioned above a user's responsibility is to provide the measurement result and the associated measurement uncertainty of a specific measurement task stemming from existing error sources. Thus far the evaluation of the uncertainty budget of samples such as step height and pitch standards, respectively, is in most cases in agreement with 1-D measurement uncertainty. To fulfill the increasing demands for 3-D measurements, it is necessary to evaluate the 3-D measurement uncertainty using a 3-D uncertainty budget. The basis of the VMM application is the existence of two closed, time-independent metrological chains in which the starting points are two distance vectors $\vec{r}_{Bb}$ and $\vec{r}_{Aa}$ derived from the common origin. The resulting vector between them is the vectorial measurement result $\vec{r}_M$. The vector $\vec{r}_{ab}$ takes into account the potential drift during the measurement (eq. 1).

$$\vec{r}_M = \vec{r}_{Bb} - \vec{r}_{Aa} - \vec{r}_{ab} = \begin{pmatrix} x_M \\ y_M \\ z_M \end{pmatrix} \quad (1)$$

The measurement result is the norm (eq. 2).

$$|\vec{r}_M| = \sqrt{x_M^2 + y_M^2 + z_M^2} \quad (2)$$

Eq. 1 and 2 are essential to calculate the measurement result, but a qualified statement to the associated measurement uncertainty is not possible yet. It is indispensable to substitute the distance vectors by sub vectors to describe the different influences on the measurement. Therefore, the metrological chains are established by substituting the distance vectors through the possible error sources $\vec{r}_i$ in a vectorized form (eq. 3).

$$\vec{r}_{Bb,Aa} = \sum_i^n \vec{r}_i \quad (3)$$

The FIGURE 2 shows the metrological chain in one touch point including the sub vectors to determine the influences on the measurement. Based on this illustration the measurement result $\vec{r}_M$ as the difference between two touching points (eq. 1) is provided by:

$$\begin{aligned} \vec{r}_M &= \vec{r}_{Pb} - \vec{r}_{Pa} + \sum_i \vec{r}_{FRai} - \sum_i \vec{r}_{FRbi} \\ &- \vec{r}_{IRa} + \vec{r}_{IRb} - \vec{r}_{IMa} + \vec{r}_{IMb} - \vec{r}_{cosa} + \vec{r}_{cosb} \\ &- \vec{r}_{Abbea} + \vec{r}_{Abbeb} - \vec{r}_{Mlea} + \vec{r}_{Mleb} + \vec{r}_{ab} \end{aligned} \quad (4)$$

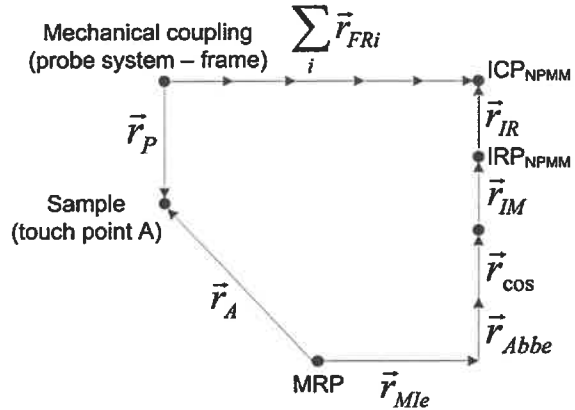


FIGURE 2. Metrological chain (touch point A).

The sub vectors (eq. 3) of the metrological chain are described in the TABLE 1 in which the interesting vector of this contribution is the cosine-error component.

TABLE 1. Sub vectors of the model.

$\vec{r}_p$	Probe system vector
$\vec{r}_{FRi}$	NPMM frame vector
$\vec{r}_{IR}$	Interferometer reference beam vector
$\vec{r}_{IM}$	Interferometer measuring beam vector
$\vec{r}_{cos}$	Cosine-error vector
$\vec{r}_{Abbe}$	Abbe-error vector
$\vec{r}_{Mle}$	Corner mirror error vector
ICP	Interferometer Center Point
IRP	Interferometer Reference Point
MRP	Mirror Reference Point

The combined uncertainty of the norm of the distance vector $u_c(|\vec{r}_M|)$ is in accordance with the GUM provided by

$$\begin{aligned} u_c(|\vec{r}_M|) &= \sqrt{\frac{(x_M * u(x_M))^2 + (y_M * u(y_M))^2 + (z_M * u(z_M))^2}{x_M^2 + y_M^2 + z_M^2}} \end{aligned} \quad (5)$$

in which the uncertainties of the measured coordinates $u(x_M), u(y_M), u(z_M)$ are derived from the uncertainty vector $\vec{u}_c(\vec{r}_M)$ [6].

COSINE ERROR VECTOR

An error source to be considered and investigated is the cosine error

$$\Delta \vec{r}_{cos} = \vec{r}_{cosb} - \vec{r}_{cosa},$$

which is inherent to interferometric displacement measurements [7]. In general, the cosine error

arises when there is an angular misalignment between the desired measurement line and the current measurement line. Because of the fact that the measurement result is a difference measurement between two touching points, the cosine error only occurs if the values of the angular misalignment between the two touching points are different.

As described before, the NPMM acts in the sample scanning mode in which the sample is positioned on the corner mirror and moved by a combination of Lorentz actuators (voice coils) and linear guides in all three measuring axes. Also, even though the x- and y-movements of the stage are monitored and actively controlled by the angle sensors as well as the laser interferometers (FIGURE 1), errors may arise due to stage tilt during the measurement. Another reason for this remaining uncertainty, the cosine error respectively, is the adjustment error of the interferometer beam and the motion axis, which means that the beams are not completely aligned to the axis of motion or perpendicular to the surfaces of the corner mirror. In addition, the flatness deviation of the corner mirror is also a factor. Although there is a correction polynomial for compensation, several approaches are done to analyze this error source [8].

To sum up, misalignment (non-perpendicularity, φ_{MA}), imperfect flatness of the corner mirror surface (φ_{FL}) and stage tilt (φ_{TI}) cause the cosine error (Δx_{cos}^2). The FIGURE 3 shows the cosine error of the line of measurement of the x-axis schematically. Within the measurement the corner mirror M is moved. This movement is detected by the x-interferometer. This measuring device consists of the Interferometer Center Point (ICP), the Interferometer Reference Point (IRP) and the diaphragm D where the reflected waves of the measuring beam interfere. The measured length is the difference between the distance $\overline{M-ICP}$ (not shown in the illustration) and the distance x_{IR} between ICP and IRP. The term x_{FR} shows the metrological frame and x_{ICD} the distance between the diaphragm and ICP. Further, the realistic case is that the corner mirror is tilted by the angle φ_{TI} . In detail, a misalignment of the measuring beam occurs because of the yaw-angle φ_{TIZ} and the pitch-angle φ_{TIY} . Referring to the illustrated x-axis, the aforementioned tilt angles lead to the so-called "geometrical cosine error" Δx_{cos-M} at the corner mirror. Next to this specified contribution to the cosine error exists also the so-called "optical cosine error" Δx_{cos-D} at the diaphragm of the interferometer. Therefore, the reasons for this

are the already mentioned existing angles due to misalignment of the measuring beam, imperfect flatness of the corner mirror and again the stage tilting. This leads to the fact that the emitted and the reflected measuring beams are not parallel. The total error angle of the "optical cosine error" follows summation.

$$\varphi = \varphi_{TI} + \varphi_{MA} + \varphi_{FL} \quad (6)$$

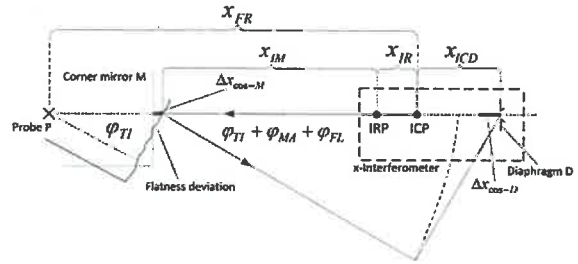


FIGURE 3. Schematic illustration of the cosine error (x-axis).

Geometrical Cosine Error

The geometrical cosine error Δx_{cos-M} is derived by following equations and the assumption that the quantity of the tilt angle is marginal.

$$\Delta x_M^z = \overline{MP}(1 - \cos(\varphi_{TIZ})) \approx \frac{\overline{MP}}{2} \varphi_{TIZ}^2 \quad (7)$$

$$\Delta x_M^y = \overline{MP}(1 - \cos(\varphi_{TIY})) \approx \frac{\overline{MP}}{2} \varphi_{TIY}^2 \quad (8)$$

Superposition leads to:

$$\Delta x_M = \overline{MP} \left(\frac{\varphi_{TIZ}^2 + \varphi_{TIY}^2}{2} \right) \quad (9)$$

The distance $\overline{MP} = x_{FR} - x_{IR} - x_{IM}$ and the error angle in vectorized form are:

$$\Delta \vec{x}_{cos-M} = \begin{bmatrix} (x_{MP}) \left(\frac{\varphi_{TIY}^2 + \varphi_{TIZ}^2}{2} \right) \\ (y_{MP}) \left(\frac{\varphi_{TIZ}^2 + \varphi_{TIY}^2}{2} \right) \\ (z_{MP}) \left(\frac{\varphi_{TIZ}^2 + \varphi_{TIY}^2}{2} \right) \end{bmatrix} \quad (10)$$

Optical Cosine Error

Similar to the geometrical cosine error, the optical cosine error for one measuring axis is defined as following:

$$\Delta x_D^z = \overline{MD}(\cos^2(\varphi_z) - 1) \approx -\overline{MD}\varphi_z^2 \quad (11)$$

$$\Delta x_D^y = \overline{MD}(\cos^2(\varphi_y) - 1) \approx -\overline{MD}\varphi_y^2 \quad (12)$$

Again after superposition it gets to:

$$\Delta x_D = -\overline{MD}(\varphi_z^2 + \varphi_y^2) \quad (13)$$

Finally, the cosine error at the diaphragm of the interferometer is provided by following matrix.

$$\Delta \vec{r}_{\cos-D} = - \begin{bmatrix} (x_{IM} + x_{IR} + x_{ICD})(\varphi_y^2 + \varphi_z^2) \\ (y_{IM} + y_{IR} + y_{ICD})(\varphi_z^2 + \varphi_x^2) \\ (z_{IM} + z_{IR} + z_{ICD})(\varphi_x^2 + \varphi_y^2) \end{bmatrix} \quad (14)$$

SIMULATION AND RESULT

A common and accepted method to analyze the quality of a measurement is the usage of the GUM uncertainty framework. It is based on the law of propagation of uncertainties and the assumption that the output quantity follows a Gaussian- or a scaled and shifted t-distribution. Additionally, the model is linearized by a first-order Taylor series approximation. Whenever these assumptions do not fulfill the postulated requirements, other methods like numerical or analytical evaluation have to be considered. One numerical solution is provided by the Monte Carlo Method (MCM). The main advantages of that is the exact analysis of linear, partly nonlinear and nonlinear models. Contrary to the propagation of uncertainties, MCM propagates probability distributions to derive an empirical distribution for the measurand which is essential to get the desired statistical values like the mean value as the best estimate or the standard deviation as the measurement uncertainty. The most difficult task is it to evaluate the uncertain quantities by the right probability distribution. In this contribution the uncertain quantities are the error angles as well as the geometrical lengths. The former ones are derived by assumption or worst case estimations. Further there are experimental setups in progress to measure them in practice. The latter ones are influenced by the coefficient of thermal expansion of the used materials and manufacturing tolerance of the used parts.

CONCLUSION

Besides a strategy of using a vectorial modular model to express the 3-D uncertainty budget this contribution also shows a possible analysis of the cosine error which is unavoidable in regard to interferometric displacement measurements in order to satisfy the high requirements set to nanocoordinate measuring devices like the NPM. On one hand, the first attempt divides the cosine error into the "optical cosine error" due to flatness deviations, non-perpendicularity and stage tilt yielding in a non-aligning emitted and reflected measuring beam, and on the other hand in the "geometrical cosine error" which arises only because of the tilt of the corner mir-

ror. The resulting uncertainties have to be calculated by means of the Monte Carlo method.

ACKNOWLEDGEMENT

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PROGRESS IN DEVELOPMENT OF AN SI-TRACEABLE PRECISION NANOINDENTATION PLATFORM

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INTRODUCTION

This paper will focus on the ongoing development of a vacuum compatible nanoindentation platform directly traceable to *Système international d'unités* (SI).

To achieve sub-nanometer performance, components of this indenter are designed to be highly symmetric and, where possible, monolithic. Incorporation of an ultra-stable laser interferometer system and state-of-the-art sensors and actuators provides sub-nanometer control and micronewton force sensing coupled with real-time signal processing for data acquisition and analysis.

OVERVIEW

This paper will focus on the performance characterization of the sub-system components of the instrument. To achieve traceability it is necessary to incorporate only predictable sources of uncertainties by utilizing a design in which all of these sub-components can be quantitatively characterized. Key areas that will be discussed include:

- Assembly and performance characterization of a novel active nulling sensor used in the creation of a reference location for indentation depth measurement that is directly on the sample surface. This mechanism will eliminate the need for instrument compliance characterization. Moreover, it creates a much-shortened metrology loop that is almost entirely separated from the force loop, does not preload the sample surface and has known

compliance where the two loops coincide.

- An evaluation of the sensitivity and long-term stability of the indenter's force and displacement transducers, including the characterization of noise sources and an analysis of the indenter's dynamic behavior and how that affects its performance.
- Comparison of indentation force-displacement data with and without surface referencing.
- General overview of the vacuum system and eddy-current-damped vibration isolation platform used to support the instrument.

Emphasis will be on experimental methods for the traceable calibration of the instrument subsystem components. The construction of a complete error budget that incorporates additional uncertainties associated with integration of these subsystem components into the complete instrument will also be discussed.

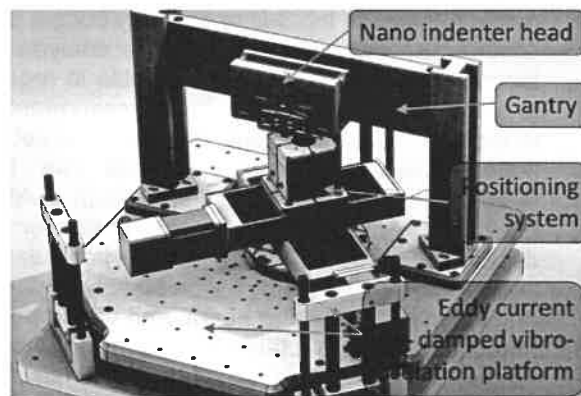


Figure 1 Nanoindentation system overview

DESIGN AND DEVELOPMENT OF SMALL-SIZED STEPPER MOTORS

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BACKGROUND

The development of small-dimension systems represents an important technological support for the scientific and industrial fields [1, 2].

There are many applications of these kind of systems such as in the medical industry, where the development of noninvasive surgery devices are required; in the metal-mechanics industry, where the necessities are focused on the manufacture of small parts like screws, gears, shafts, etc. [3]; in the educational field, where the small-sized mechatronic systems allow the teaching of CNC systems, control systems and manufacturing techniques, among others.

With the growth of this area, the development of actuators to automate these systems was required in order to minimize errors from manual control and to increase their performance characteristics, such as the precision. There are some alternatives of actuators based on electrostrictive, hydraulic, pneumatic, shape memory alloys, and electromagnetic principles; a lot of them are expensive and require the development of complex control systems.

Stepper Motors represent an easy and low-cost alternative to open-loop control systems. Its easy implementation proves to be an advantage against other alternatives such as DC Motors. On the other hand, some disadvantages of this kind of actuators are related with its low torque output and its relative low-speed operation. In this work, the development of small-dimension stepper motors is presented. Two configurations are proposed: the first one is based on the permanent magnet stepper motors principle, and the second one is based on the Lorentz force principle.

Some of the actuators developed have been employed in the automation of small-dimensions systems such as micromachine tools. An example of an automated micromachine tool [3] is shown in Figure 1.

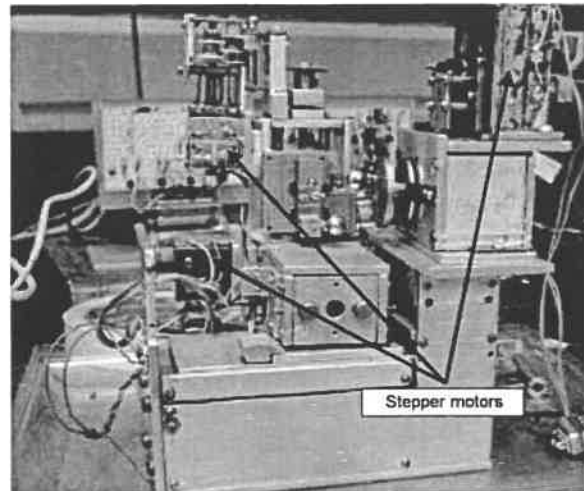


FIGURE 1. Stepper motors employed in a first-generation micromachine tool.

The stepper motors shown in Figure 1 have the following characteristics: Current consumption of 0.5 A, 4 steps per revolution, approximately 4.3 mNm for the axes' actuators and 7 mNm for the spindle actuator. The overall dimensions of the axes' actuators are 25x25x40 mm and of the spindle actuator are 40x40x20 mm.

Nowadays, with the help of CAD-CAM systems and CNC systems, there are new designs of stepper motors with better characteristics in order to employ them in the design of novel micromechanical systems. The challenge is to obtain the required efficiency of the applications because some works report that the efficiency of the electromagnetic actuators can decrease with

their size [4]. In order to exemplify the achievements obtained, the development of some small-dimension stepper motors and their functional characteristics are presented.

SMALL-DIMENSION STEPPER MOTORS DESIGN

In this work, the development of actuators with dimensions under 30x30x30 mm is considered. Two types of stepper motors are proposed: the first one employs the principle of the permanent magnet stepper motors with a double stator configuration [5] in order to obtain better characteristics of torque, and the second one is based on force generated in the interaction between a magnetic field of a magnet over a current in a wire called Lorentz force with the same configuration of the first motor.

Double Stator PM Stepper Motor

Two stators and a rotor between them compose this motor. In each stator, four coils placed equidistantly from the center of the stator with a relationship of 90° between them. The coils are wired in order to obtain two phases. The rotor is conformed by a disc with magnetic inserts. The number of the inserts determines the resolution of the motor. This configuration increases the efficiency because there are less magnetic losses. Figure 2 shows a schematic of this motor.

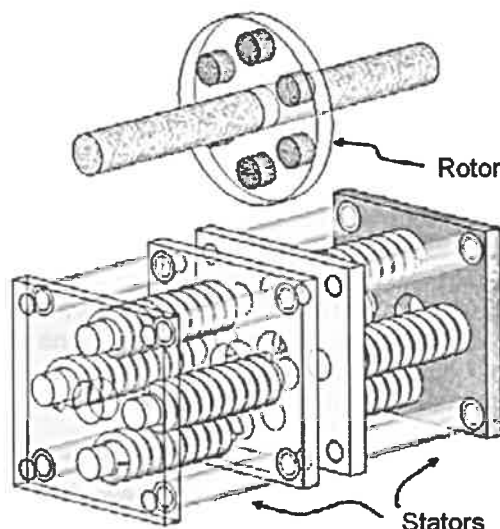


FIGURE 2. Double stator stepper motor.

Double Stator Lorentz Force Stepper Motor

This motor is based on the force exerted on a charged particle by electric and magnetic fields. When a conductive straight wire carrying a current is affected by a perpendicular constant magnetic field, the force is obtained by

$$F = L i \times B \quad (1)$$

Where L is the wire length, i is the total current and B is the magnetic field.

The proposed stepper motor is conformed, as the PM motor explained before, by a rotor with magnetic inserts arranged equidistantly from the center. The rotor is placed between two stators with four coils each one and are wired in order to conform two phases. When a current is presented, a force is generated and the rotor moves to the equilibrium position of the electromagnetic array. Figure 3 shows a scheme of the proposed design.

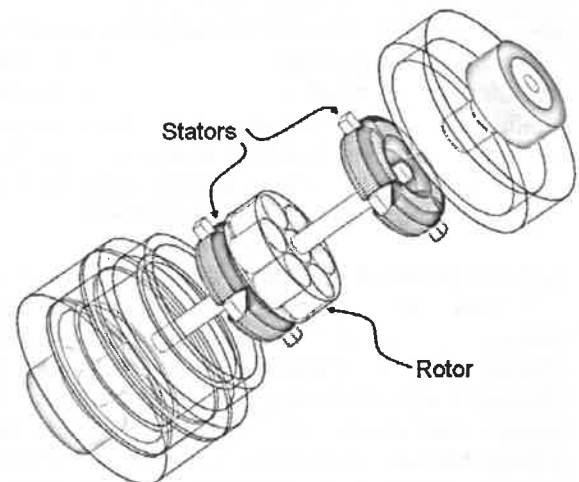
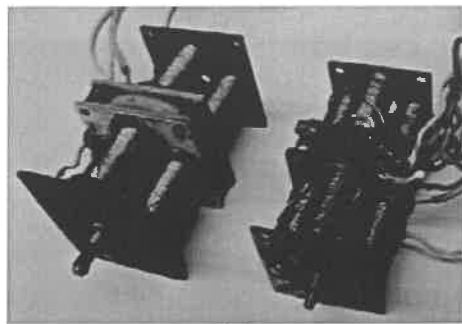


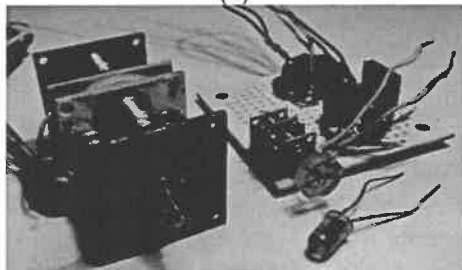
FIGURE 3. Double stator Lorentz force stepper motor.

Stepper Motors Development

In order to show the functioning of the motors explained before, three motors of each configuration were fabricated. In the case of the double stator PM stepper motor, the overall sizes of the motors were 25x25x43 mm with 28 steps per revolution, 20x20x41 with 20 steps per revolution and 10x10x12 mm with 20 steps per revolution, while for the double stator Lorentz force stepper motor were ø32x25mm with 26 steps per revolution, ø25x21mm with 12 steps per revolution and ø10x11mm with 12 steps per revolution. In Figures 4 and 5, the fabricated motors are shown.



(a)



(b)

FIGURE 4. Double stator PM stepper motors. (a) The 25x25x43mm motor compared with the 20x20x41mm motor; (b) the 25x25x43mm motor compared with the 10x10x12 mm motor.

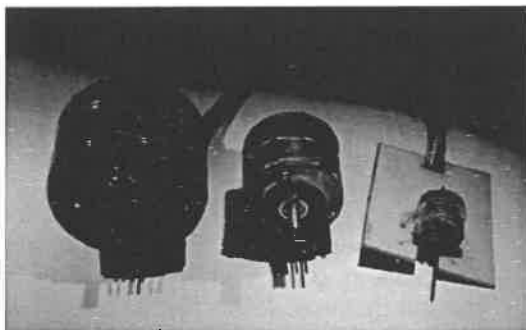


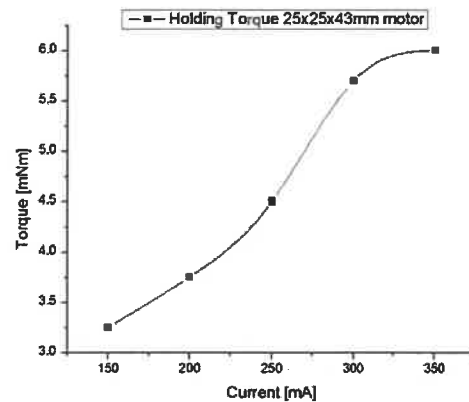
FIGURE 5. Double stator stepper motors.

RESULTS

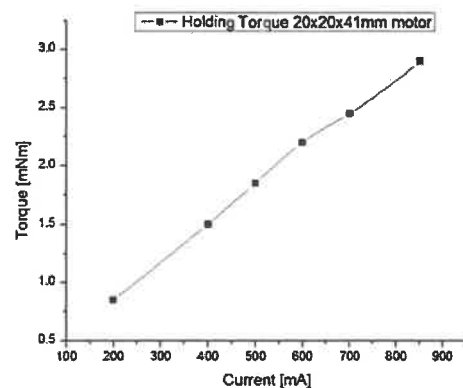
In order to evaluate the behavior of the fabricated motors, the holding torque was characterized. This parameter corresponds to the torque required for a motor to keep on in a fixed position when a phase is energized. Figures 5 and 6 show the experimental holding torque for the motors developed.

In order to compare the performance of both designs, a comparison was performed taking into account the torque/volume ratio for the middle sized motor of each design. For the 20x20x41mm motor, the torque/volume ratio was $0.18 \mu\text{Nm}/\text{mm}^3$, while the torque/volume

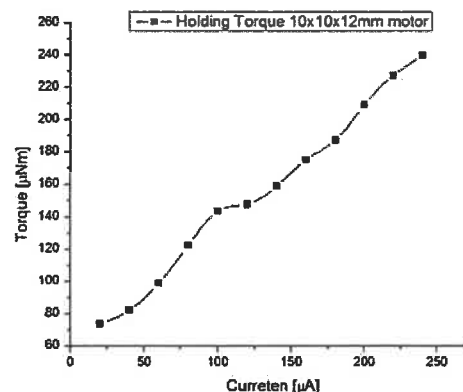
ratio for the $\varnothing 25 \times 21 \text{mm}$ is $0.13 \mu\text{Nm}/\text{mm}^3$. For this case, the Double stator stepper motor generates 38% more torque per volume than the Double stator Lorentz force stepper motor.



(a)



(b)



(c)

FIGURE 5. Double stator stepper motor holding torque. (a) 25x25x43mm motor; (b) 20x20x41mm motor; (c) 10x10x12 mm motor.

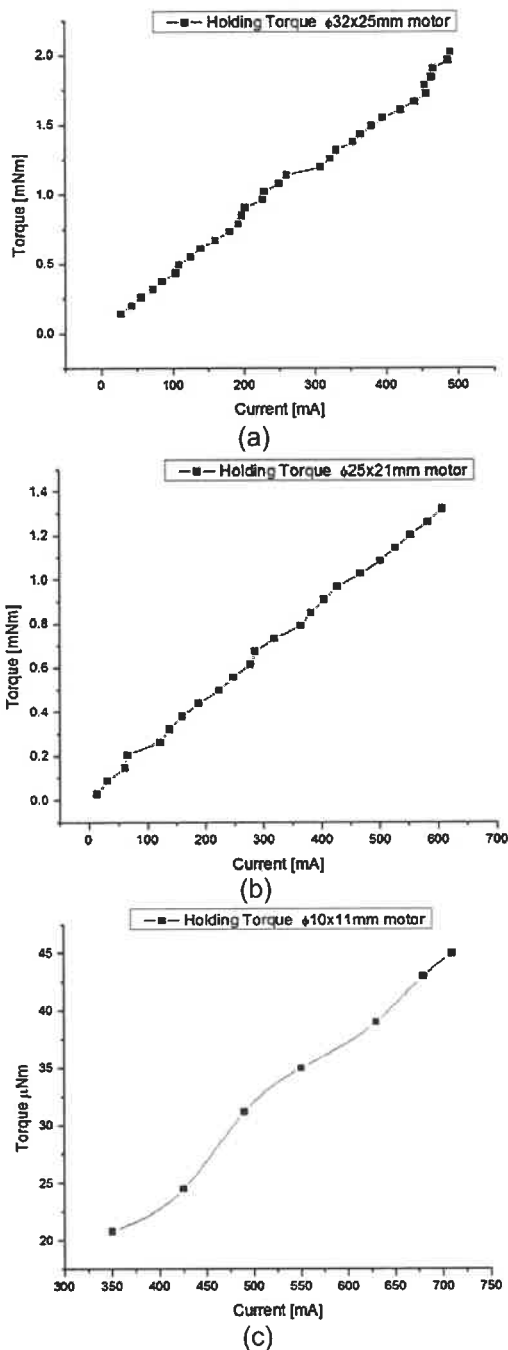


FIGURE 6. Double stator Lorentz force stepper motor holding torque. (a) $\phi 32 \times 25 \text{ mm}$ motor; (b) $\phi 25 \times 21 \text{ mm}$ motor; (c) $\phi 10 \times 11 \text{ mm}$ motor.

In order to demonstrate the applications of these actuators, double stator PM stepper motors were implemented in a micromachine tool (see Figure 7).

The micromachine tool shown in Figure 7 has overall sizes of $120 \times 160 \times 85 \text{ mm}$, resolution of

140 nm per motor step and precision better than $10 \text{ }\mu\text{m}$.

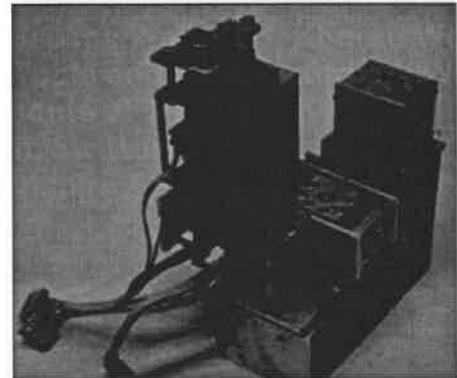


FIGURE 6. 3-axes micromachine tool.

CONCLUSIONS

The development of stepper motors for the automation of small dimension systems represents a good and low-cost alternative. The Double stator stepper motors have better torque/volume than the double stator Lorentz force stepper motors. On the other hand, the design of the double stator Lorentz force stepper motors is easier because the output force is determined by a simple equation.

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EXPERIMENTAL DETERMINATION OF NUT STIFFNESS IN BALLSCREWS

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ABSTRACT

To verify previously developed nut stiffness model of a ballscrew, an experimental setup measuring the axial deformation and the stiffness of a ballscrew has been devised. Single-nut and double-nut ballscrews are tested under several load and assembly conditions. It is confirmed that experimental values are around 80% of the mathematical nut stiffness.

KEYWORDS: Ballscrew, Body deformation, Hertz contact, Nut stiffness, Preload, Uncertainty

INTRODUCTION

Ballscrews are common positioning units applied to industrial machinery and precision machines. As the positioning accuracy of a ballscrew driven system depends upon stiffness of the leadscrew system, stiffness estimation of the ballscrew is critical for accurate positioning and design of the precision machinery. Axial stiffness of the feed drive system consists of stiffnesses of the nut, screw shaft, supported bearings, and nut bracket. Recalling that the contact deformation of the nut assembly governed by Hertz contact is affected by the groove shape, preload and thrust force as shown in Fig. 1, it has been difficult to derive an accurate mathematical model of the nut stiffness [1-2]. To formulate an accurate nut stiffness model, flexibility of the screw shaft and nut was added to the Hertz contact deformation model [3].

In this paper, to confirm and verify the previously developed mathematical model [3] of the ballscrew nut stiffness, measurement of axial deformation and stiffness is conducted on the stiffness measurement system. Single-nut and double-nut ballscrews are tested under several load and assembly conditions.

EXPERIMENTS

Fig. 1 shows cross sections of the single-nut and double-nut ballscrews in TC (Tension in the shaft and Compression in the nut: forward motion case)

and TT (Tension in the shaft and Tension in the nut: backward motion case) load conditions. Contact state is shown in Fig. 1(c).

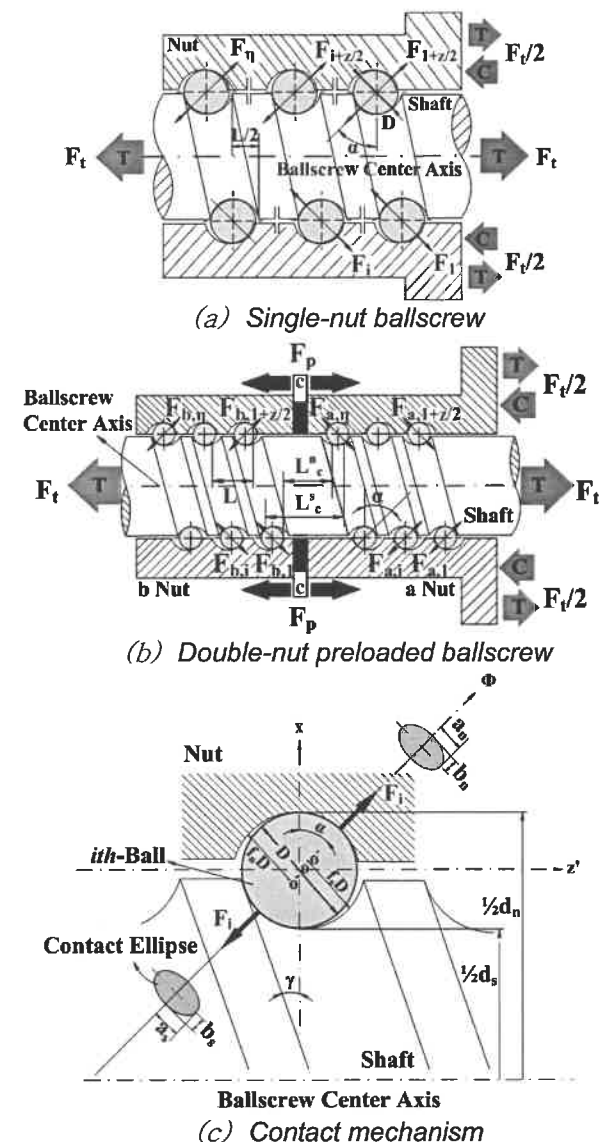


FIGURE 1. Ballscrew geometry, load conditions and contact mechanism.

D means the ball diameter, α the contact angle of the ball and grooves, γ the lead angle of the screw shaft, F_i the contact load, F_t the thrust force applied, and f_s and f_n conformity factors of the screw and nut grooves, respectively.

According to Hertzian contact theory [4], and previously developed nut stiffness model [3], mathematical modeling of ballscrew stiffness has been conducted through the contact and body deformation analysis in various load and assembly conditions. Body deformation between elements $i-1$ and i is given by,

$$\begin{aligned}\delta_{i-1,i}^s &= (\tilde{\delta}_{i-1}^s \sin \alpha_{i-1} - \tilde{\delta}_i^s \sin \alpha_i) \cdot \cos \gamma \\ &= \frac{L/2}{E_s \cdot A_s} [(F_{i-1} \sin \alpha_{i-1} - F_i \sin \alpha_i) \cdot \cos \gamma]\end{aligned}\quad (1)$$

$$\begin{aligned}\delta_{i-1,i}^n &= (\tilde{\delta}_{i-1}^n \sin \alpha_{i-1} - \tilde{\delta}_i^n \sin \alpha_i) \cdot \cos \gamma \\ &= \frac{L/2}{E_n \cdot A_n} [(F_{i-1} \sin \alpha_{i-1} - F_i \sin \alpha_i) \cdot \cos \gamma]\end{aligned}\quad (2)$$

$$\delta_{i-1,i}^B = -\delta_{i-1,i}^s \pm \delta_{i-1,i}^n \text{ for [TC(+); TT(-)]} \quad (3)$$

Total body deformation is obtained as

$$\begin{aligned}\delta_{total}^B &= -\sum_{i=2}^P \delta_{i-1,i}^s \pm \sum_{i=2}^P \delta_{i-1,i}^n \\ &\text{for [TC(+); TT(-)]}\end{aligned}\quad (4)$$

Considering contact deformation in axial direction, δ_a^H , total axial deformation is given by [3]

$$\delta_{total} = \delta_a^H + \delta_{total}^B \quad (5)$$

To verify the developed nut stiffness and deformation models, the stiffness measurement system composed of the hydraulic cylinder, two gap sensors (resolution 0.1 μ m, ONO SOKKI VT-120 with VS-011), load cell (Bongshin, 2 tons) and data acquisition system (LabView PXI-8187) shown in Fig. 2 is devised.

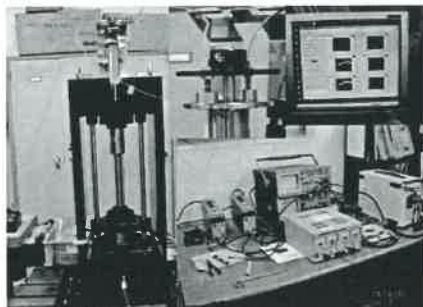


FIGURE 2. Stiffness measurement system.

The sensors are calibrated with the electronic length measuring equipment (resolution 0.1 μ m, TESATRONIC TT 20) and the Tesa's precision slide as shown in Fig. 3. Data sampling of axial deformation and thrust force are conducted simultaneously with sampling rate of 3Hz and filtered by a moving average over 9 samples. Experiments are performed according to the load and assembly conditions.

Single-nut Case

Using the previously formulated mathematical model [3], axial deformation and stiffness of NSK's single-nut ballscrew (Model: SFT2505-5) in the TT load condition (same as of CC) are computed as shown in Figs. 4 and 5. Ballscrews with similar geometry and characteristics are applied for experiments and verification of the mathematical models [3]. Corresponding to the single-nut NSK's SFT2505-5 for the simulation, THK's BNF2505-5 are selected for experimental verification. Figure 6 shows thrust force and axial deformation according to loading step, and axial deformation and stiffness versus thrust force. In the figures, different colors stand for different sensors and mean values of the sensors. Based on Figs. 4 and 5, mathematically obtained axial deformation and stiffness of SFT2505-5 are 5.8 μ m and 57Kgf/ μ m at 500Kgf thrust force. Measured deformation and stiffness are 10.6~11.4 μ m and 47~44Kgf/ μ m at 500Kgf thrust force, respectively. Comparing the experimental stiffness with the analytical value, it is around 80% of the theoretical stiffness. Fig. 7 shows stiffness measurement at the different nut location of the same ballscrew. It is 42~46Kgf/ μ m (77% of the analytical value) at 500Kgf thrust force. As the single-nut does not have preload, there is slippage due to backlash at the initial loading state during experiment. This effect generates uncertainty during experiment. However, it confirms that the previously developed stiffness model of the single-nut has 78.5% accuracy comparing with experimental results.

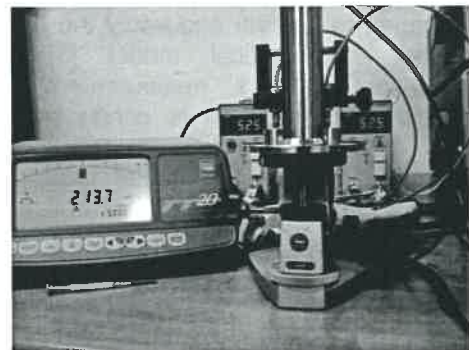


FIGURE 3. Sensor calibration with Tesa equipment.

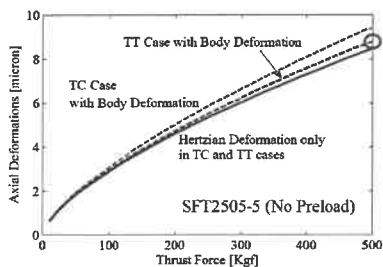


FIGURE 4. Axial deformation vs. thrust force.

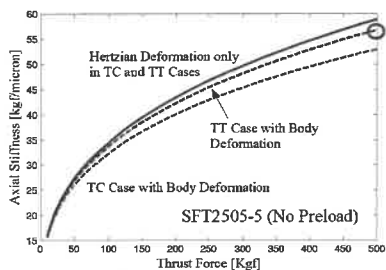


FIGURE 5. Nut stiffness vs. thrust force.

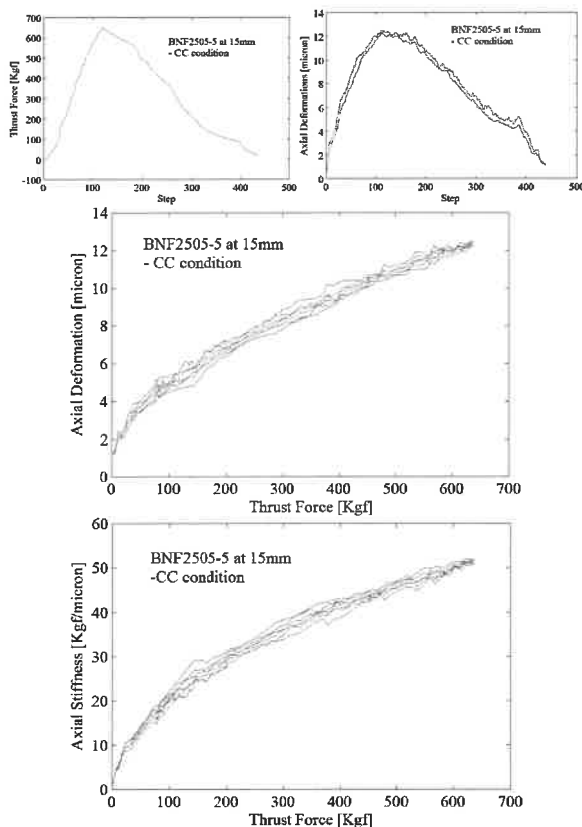


FIGURE 6. Stiffness measurement in CC load condition at 15mm location.

Double-nut Case

Double-nut ballscrews are generally preloaded to eliminate backlash and improve stiffness. Figs. 8 and 9 show analytical axial deformation and stiffness of NSK's double-nut ballscrew (Model: DFT

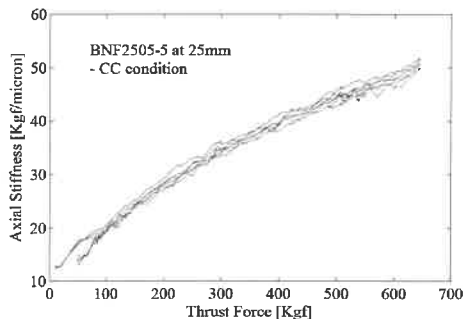


FIGURE 7. Stiffness measurement in CC load condition at 25mm location.

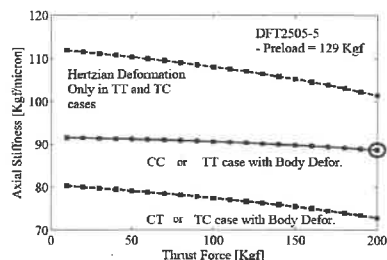


FIGURE 8. Axial deformation vs. thrust force.

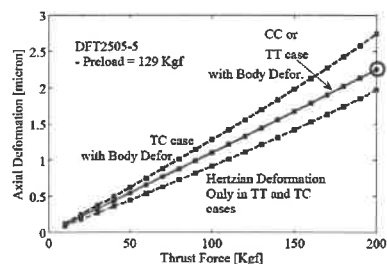


FIGURE 9. Nut stiffness vs. thrust force.

2505-2.5) with preload 129Kg in the CC load condition, which are obtained from previously derived mathematical stiffness models [3].

Corresponding to the double-nut NSK's DFT2505-5 for the simulation, THK's BNFN2505-5 are selected for experimental verification. Figure 10 shows thrust force and axial deformation according to oscillated loading step, and axial deformation and stiffness versus thrust force. During the experiment, preload is fully released at the critical load around 360Kg thrust force, and the loading continues up to 1,000Kg thrust force. Within the critical region, the stiffness decreases according to the load increase. The stiffness increases after passing the critical load. Contrary to the single-nut experiment, slippage of the loading mechanism is little and stiffness is maintained high during loading. In the figure, different colors stand for different sensors and mean values of the sensors. Based on Figs. 8 and 9, mathematically obtained axial deformation and

stiffness of DFT2505-5 with 129Kgf preload are $2.25\mu\text{m}$ and $89\text{Kgf}/\mu\text{m}$ at 200Kgf thrust force. Measured mean deformation and stiffness are $2.7\mu\text{m}$ and $75\text{Kgf}/\mu\text{m}$ at 200Kgf thrust force, respectively. It confirms that experimental stiffness is around 84% of the analytical value. In addition, THK's recommended stiffness value of BNFN 2505-5 at 170Kgf thrust force is $93\text{Kgf}/\mu\text{m}$. It is obtained as $80\text{Kgf}/\mu\text{m}$ (86% similarity) in the experiment. This 85% of estimation accuracy may be improved when the input parameters such as conformity factor, contact angle, ball diameter, Young's modulus, *etc.* of the mathematical model is furnished accurately.

Fig. 11 shows stiffness measurement result at the different nut location of the same ballscrew. It is $68\sim 78\text{Kgf}/\mu\text{m}$ (78.5% of THK's recommended value) at 170Kgf thrust force, and $70\sim 78\text{Kgf}/\mu\text{m}$ (83.1% of the mathematical model) at 200Kgf. It is confirmed that the previously developed analytic stiffness model [3] of the double-nut has 82.9% accuracy comparing with experimental results. According to Fig. 10 and 11, there is big uncertainty during the stiffness measurement in the initial and the loading after 600Kgf thrust force regions. These are due to uncertainty of sensors, clearance of the loading mechanism, and titling of the ballscrew shaft during experiments. To reduce the uncertainty, accurate input parameters for the mathematical model, the nano-level sensor, and 3 sensors located every 120° interval are required.

CONCLUSIONS

Experimental verification of the nut stiffness model of ballscrews are performed through experiments. It is confirmed that experimental stiffness of single-nut and double-nut ballscrews are around 80% of the previously developed mathematical nut stiffness model. The stiffness is estimated accurately when the input parameters such as conformity factor, contact angle, ball diameter, Young's modulus, *etc.* of the mathematical model is furnished correctly. Experimental uncertainty is able to be reduced through the clearance free mechanism of the experimental setup and the usage of 3 sensors with nano-level accuracy.

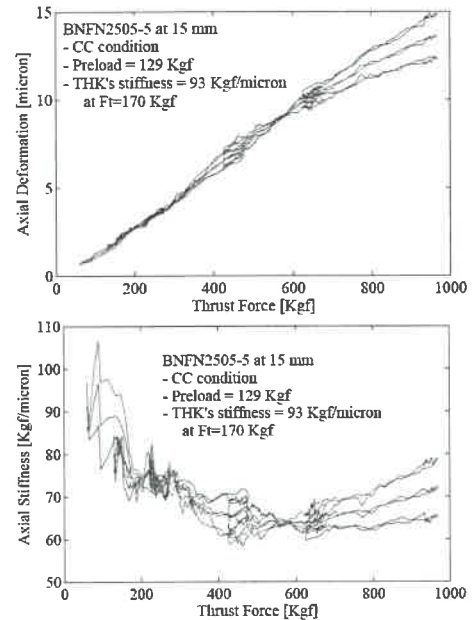
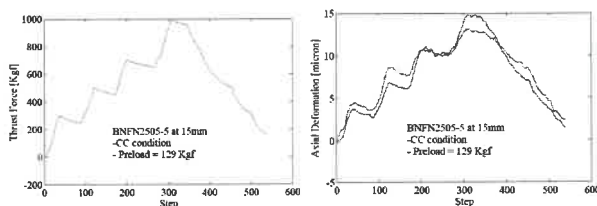


FIGURE 10. Stiffness measurement in CC load condition with 129Kgf preload at 15mm location.

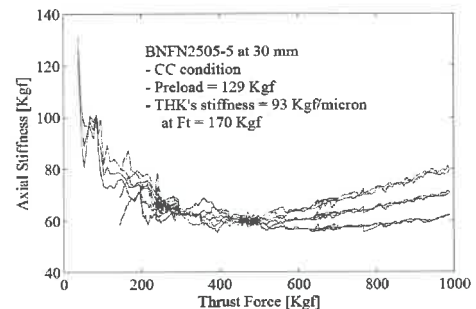


FIGURE 11. Stiffness measurement in CC load condition with 129Kgf preload at 30mm location.

ACKNOWLEDGEMENT

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THERMAL MODEL OF MACHINE TOOL FEED DRIVETRAINS

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INTRODUCTION

Thermal effects play a major role in the accuracy of machine tools [1]. Using thermo-mechanical simulation the effects of changes in the environmental temperature are often analyzed [2]. In this case the modeling is simplified, as the machine itself is not moving, and only convection has to be considered. Another important research field concerns the thermo-mechanical behavior of spindles, which can show a significant growth due to temperature increases. In [3] a thermo-mechanical model is shown, which however does not give any information on the electric losses. Linear and rotary axes also have to be considered as they generate heat due to the movement of the machine tool. In [4] different ball screw systems have been modeled and analyzed in order to reduce the effects of an elongation of the ball screw. Feed drive temperatures however had to be measured as no loss generation model for the drive exists. Along the drive train of a linear axis a lot of different machine elements, guides and feed drives e.g., have to be considered. In this work all losses and frictional forces for each machine element will be calculated and used as input for the thermal model. A special focus is put on the feed drive, which is modeled using finite differences method and then is added to the FE (finite element) model of the machine tool. The simulated losses, the current and also temperatures are compared with the values measured on a machine tool. It will be shown that the model can be used to predict temperatures, losses and power consumption of the CNC axis for a given NC path.

THERMAL NETWORK

In this section a general thermal equivalent network is derived, which can be used for most electric drives occurring in machine tools. Usually the thermal effects of electric drives are modeled using one or two body systems, where the whole drive or stator and rotor are treated as a single homogenous body. In the case of

machine tools a more detailed approach is beneficial as sometimes more information is needed.

Heat Resistances and Capacitances

The basis for the thermal network is given by [5] which was created for totally enclosed fan cooled asynchronous motors. Fig. 1 shows the general thermal network which was derived to model all types of electric drives. The thermal network basically represents a radial cut of an electric drive. In the center there is the shaft and the rotor. Heat resistances and capacitances for a thermal equivalent network can be calculated quite efficiently. Considering a cutaway of a cylinder one can define radial, circumferential and axial resistances, which are shown in many textbooks. Calculation of more complicated heat resistances will be shown in the following sections.

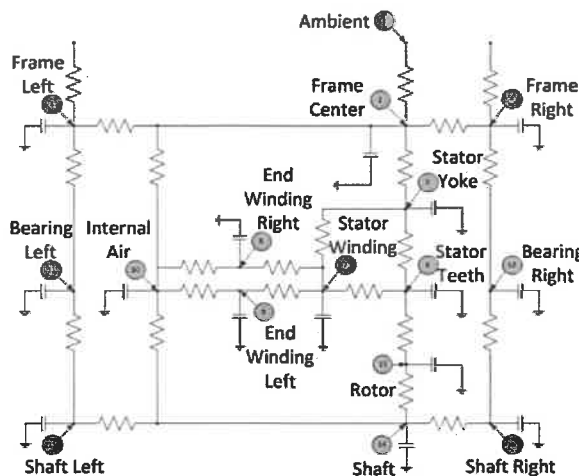


FIGURE 1. Finite difference representation of the feed drive, with its nodes, heat resistances and capacitances.

Air Gap

Between the stator and rotor there is a small air gap, through which heat can flow in the radial direction. The heat resistance of this air gap is

calculated using the modified Taylor number Ta_m Eq. 1 to 3, from [6].

$$Ta_m = \frac{\omega^2 d \delta^3}{\nu^2} \quad (1)$$

$$Nu = 2, Ta_m < 1740 \quad (2)$$

$$Nu = 0.409 Ta_m^{0.241} - 137 Ta_m^{-0.75}, Ta_m \geq 1740$$

$$\alpha = \frac{Nu \lambda}{2 \delta} \quad (3)$$

It shall be noted that the heat conductivity and viscosity of air are temperature dependent. In a very accurate examination the size of the air gap would also be temperature dependent, due to the thermal expansion of the rotor and shaft.

Insulation

Between the stator copper and the iron there is also insulation. The insulation is very thin and therefore is not considered as an independent node in the thermal network. However the materials used have very low heat conductivity. Therefore they have to be included in the heat resistance between the copper and iron. The thickness of the insulation layer is not exactly known, but often a thickness of 1mm is given. Here a heat conductivity value for an insulation material, in this case Nomex®, is used.

Housing

A variety of different housing types exist, each with a different convection coefficient. Convection α for different housing types is given in [7] as Eq. 4 with different coefficients k_1 and k_2 , shown in Tab 7.2.

$$\alpha = k_1 \left(\frac{\Delta T}{D} \right)^{k_2} \quad (4)$$

where D is the housing diameter and ΔT is the temperature difference between housing and ambient.

TABLE 1. Coefficients for different housing types

Housing Type	k1	k2
Cylindrical	1.30	0.25
Square	1.078	0.25
Standard	1.176	0.25
Radial Finned	0.748	0.33

Losses

Heat generated inside an electric drive can be divided into three types: Copper losses due to

the current which is necessary to create the magnetic field, losses inside the iron which occur due to the changes in the magnetic field, as well as eddy-current losses and finally friction losses occur in the bearing and also inside the air gap. Usually losses inside the air gap are small enough to be neglected.

Copper Losses

The formula for calculating copper losses, which are caused by the resistance and current flowing through a conductor, is well-known. In case of a electric drive with three phases the formula is given by Eq. 5.

$$P_{copper} = 3R(T)I^2 \quad (5)$$

For modeling the copper losses the hardest part is to get appropriate values for the resistance of a phase, which is temperature dependent, and the actual current. Values for the resistance can usually be found in the data sheet of the electric drive. Calculation of the stator current in permanent magnet synchronous drives can be achieved using the current-torque constant [8].

$$M \approx k_t(T)I \quad (6)$$

The strength of rare earth magnets also is temperature dependance. Feed drives usually have small enough temperature changes, to consider k_T being constant. However, it can have an impact, when temperature increases are high, as e.g. on vertical axes.

Iron Losses

Giving an exact representation of the iron losses is a complicated procedure and is influenced by many parameters, e.g. the geometry of the iron core. Since these parameters are not known in detail, the iron losses can be approximately calculated using Eq. 7 [8]. The exponent depends on the internal geometry of the iron.

$$P_i = P_{i,rated} \frac{f^{1.5...2}}{f_{rated}} \quad (7)$$

The iron losses at rated frequency $P_{i,rated}$ have to be known. Frequency dependence is given by the current frequency f and the rated frequency f_{rated} .

Bearing Losses

In the data sheets one can usually find a value for the static friction of the feed drive. In a first step this value is sufficient to model the bearing losses. The static friction is given by the friction of both bearings. Therefore the bearing losses P_b are calculated according to Eq. 8.

$$P_b = \frac{M_{static}}{2} \omega \quad (8)$$

This simple model will not be sufficient for a spindle since much higher rotational speeds will be achieved and centrifugal forces significantly influence the load and path of the rolling bodies.

LOSSES OF THE WHOLE DRIVE TRAIN

Besides in the feed drive, losses also occur in the guideway, the ball screw and the thrust bearing. Apart from dynamic forces, caused by acceleration and deceleration, frictional forces in these machine elements have to be overcome by the feed drive. Appropriate models for these machine elements, considering load and speed dependency had to be implemented here. Therefore it is necessary to calculate the load and the speed for each machine element individually. These models are integrated into a global model of the drive train, which is used to calculate the frictional forces acting on each machine element of the guideway under load. The elements in the kinematic chain have to carry the frictional forces of the previous elements. This is shown in Fig 2, where for each element, it is symbolized which friction forces are carried by it. The ballscrew for example has to carry the friction forces of the guideways, indicated in the grey circle. Besides the friction forces, the inertial forces of course are also considered.

INTEGRATION INTO FEA

The losses which are calculated based on the machine motion in the previous section are then used in a thermo-mechanical finite element analysis to calculate both the temperature and deformation of the machine tool. Since the feed drive is modeled using finite differences and not finite elements, as opposed to the rest of the machine structure, the feed drives are not explicitly geometrically modeled, as can be seen in Fig. 3. It can be seen that the whole structure of the machine tool is modeled geometrically, but for the feed drive only the mounting flange is visible. To integrate the thermal network of the

feed drive, the FEA equation system is expanded by degrees of freedom which are needed for the equation system of the feed drive. The system of equations of the feed drive is then coupled to the finite element equation system by the appropriate heat resistance of the mounting.

SIMULATION AND MEASUREMENTS

Measurements on machine tools have been conducted while performing a pendular movement of a linear axis (according to ISO 230-3). This movement then is recreated in the FE simulation of the machine tool. The simulated temperature values of the machine elements are compared with the measurements, as well as the calculated current values of the feed drive.

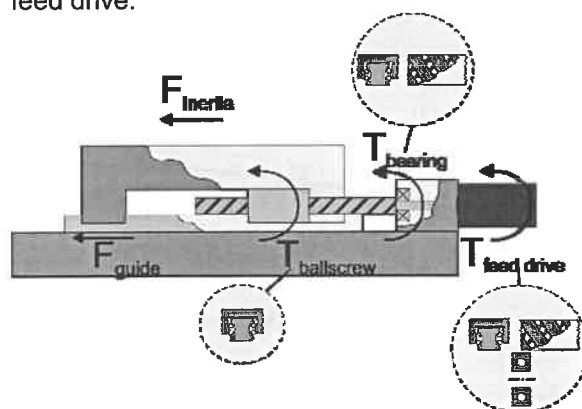


FIGURE 2. Example for load calculation: Inertia and friction forces at the different machine elements and the total load acting on the feed drive

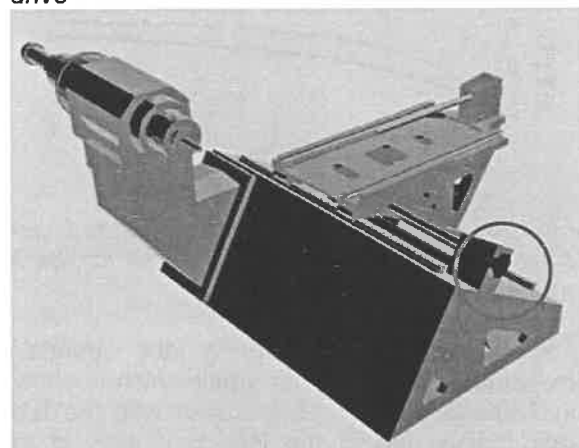


FIGURE 3. Part of the FEA model of the machine tool and flange (inside red circle) where the finite difference system of the feed drive (X-axis) is connected to.

The pendular movement was performed on two different axes on the machine tool. The

measurements on each axis have been made with various feed rates in order to validate the model over the whole speed range of the axis. This is necessary since at different speeds and different torques the main source of heat losses changes.

In Fig 4 the comparison of the measured and simulated temperature values at the motor flange can be seen for three different feed rates of the X-axis, 500mm/min, 1500mm/min and 3000mm/min respectively.

Even though only parameters given by the manufacturer's documentation have been used, it can be seen that the correlation between the simulation and measurement is good. The method is desirable in order to predict the thermal behavior of machine tools, making subsequent parameter identification almost obsolete.

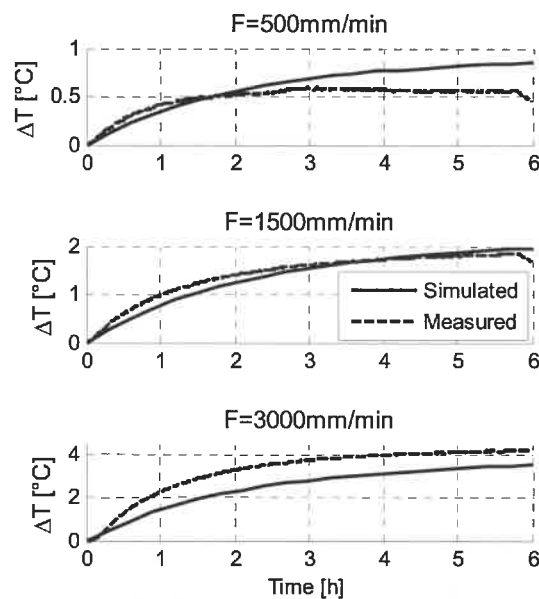


FIGURE 4. Comparison of measured and simulated motor flange temperatures for the X-axis at different feed rates.

The overall losses during the pendular movements are all rather small which is shown in Table 2. The losses increase with the feed rate mainly due to the increased loss in the bearing and also to a slight extent the iron losses. The required current, and therefore copper losses, are decreased with higher speeds because the torque curve of the whole drive train resembles the Stribeck curve. If higher feed rates would have been measured,

the torque, and as consequence the current, would increase again.

TABLE 2. Coefficients for different feed rates

Feed Rate [mm/min]	Loss [W]	Current [A]
500	1.30	0.46
1500	2.58	0.37
3000	4.75	0.33

CONCLUSIONS

It has been shown that using the model for the feed drive train, good agreements between the simulated and measurement temperature values at different feed rates can be achieved. Further validation using force measurements will be performed, in order to compare the simulated and measured load on the feed drive.

ACKNOWLEDGEMENT

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LOAD BEARING CAPACITY AND STIFFNESS OF POROUS JOURNAL GAS BEARING ELEMENTS IN A VACUUM ENVIRONMENT

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ABSTRACT

The properties of porous journal gas bearing elements change, when operated under vacuum conditions. Within this paper, the experimental obtained load bearing capacity and stiffness of gas bushings are compared, using the designed setup at normal atmosphere and at vacuum. The results are essential for the development of compact, high vacuum compatible precision positioning stages.

INTRODUCTION

Many positioning processes, especially within the semiconductor industry or for high-precise measurements have to be performed within a vacuum environment since disturbing effects like absorption or the change of the refraction index have to be eliminated. If the accuracy of those positioning processes have to be within the nanometer range the usage of gas bearing guides is beneficial since they operate with negligible friction and without stick-slip.

Gas bearing elements within a vacuum environment can be operated if certain measures are undertaken. The commonly used method to keep the vacuum quality is to exhaust the supplied gas before reaching the vacuum chamber.

Different kind of seal systems can be found in the literature, but most of them have a similar setup with a plurality of exhaust grooves and seal gaps comprising a high flow resistance [1] - [5]. The first exhaustion groove can be opened to normal atmosphere where the following grooves are connected to vacuum pumps [1]. This setup results in known properties of the gas bearing since they are surrounded by normal atmosphere. The disadvantage is, that the pressure in front of the first seal gap is an ambient pressure as well, resulting in a comparably high leakage rate through the first seal gap.

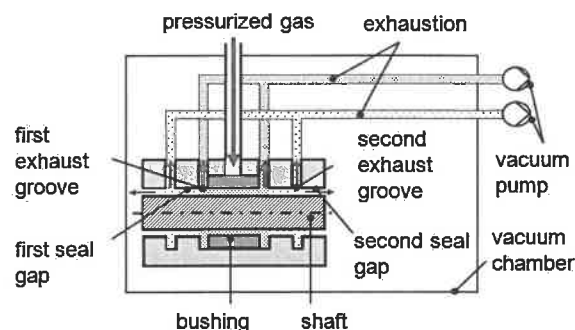


FIGURE 1. Schematic setup of a vacuum compatible gas bearing guide with two-stage exhaust system.

A known modification [3]-[5] of the setup is shown in FIGURE 1. All exhaust grooves are connected to vacuum pumps whereby the pressure in front of the first seal gap is reduced to a low to medium vacuum. Hence, the leakage rate through the first seal gap is reduced as well. This results in a better vacuum level within the vacuum chamber using a compact seal system. An effect of this reduced surrounding pressure is the change of the properties of the porous journal gas bearing elements (hereafter referred to as bushings).

For the appropriate design of a positioning system using bushings and the described exhaust method, the character of the change of the load capacity and the stiffness has to be known.

FUNDAMENTALS

Flow regimes

The common classification into the molecular, the intermediate and the viscous flow is based on the Knudsen number Kn that is the ratio of the mean free path of the gas λ and the characteristic length of the element, e.g. the height of the bearing gap h . The according pressure p is for constant boundary conditions inverse proportional to λ with a constant factor c . Presuming a temperature of 273 K and the usage of nitrogen, this factor becomes $c = 6.376 \cdot 10^{-3} \frac{kg}{s^2}$.

$$Kn = \frac{\lambda}{h} = \frac{c}{h p} \quad (1)$$

A supply pressure for the bushing of $p_s = 300,000 Pa$ within the bearing gap of several microns generates low Knudsen numbers ($Kn < 0,01$) which accounts for viscous flow. By reducing the surrounding pressure within the first exhaust groove to a low to medium vacuum, a transition to intermediate ($0,01 < Kn < 0,5$) and molecular flow ($Kn > 0,5$) can occur.

For different flow regimes, different analytical approaches for the calculation of the pressure distribution within the bearing gap have to be used, which can be found in the literature [6]. This transition causes a change of the pressure distribution over the bearing surface and hence different properties of the bushing during the change from normal atmosphere to a vacuum environment.

Structure of investigated bushings

The investigated bushing from NewWay air bearings® has an inner radius r_B of 20.012 mm $\pm$ 0.001 mm and a length l of 76.20 mm. The used shaft has an according outer radius r_S 20.000 mm $\pm$ 0.004 mm. In FIGURE 2, parameters and the used coordinate system for the bearing gap between the bushing and the shaft is shown.

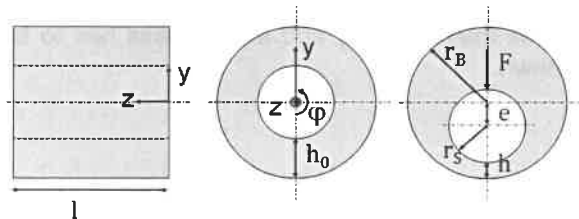


FIGURE 2. Schematic view of bearing gap with coordinate system and parameters

The uniform bearing gap $h_0 = r_B - r_S$ between the bushing and the shaft is valid as long as no radial force interacts with the system. If a radial force is applied, the bearing gap becomes a function of the resulting eccentricity e and the position along φ .

$$h = h_0 - e \cdot \cos \varphi \quad (2)$$

The minimal gap height h lies within the same plane as the force vector.

Properties

The properties which are investigated within this paper are the load bearing capacity and the stiffness of the bushing.

The load bearing capacity F is the force that can be supported by the bushing at a certain gap height. It can be defined by integrating the product of the pressure and the projected radius over the whole bearing surface.

$$F = \int_0^{2\pi} \int_0^l p \cos \varphi r_S dz d\varphi \quad (3)$$

Since the pressure distribution within the bearing gap changes with the gap height, the load bearing capacity is a function of the gap height. To experimentally obtain the load bearing capacity, a known force can be applied to the bushing-shaft setup while measuring the resulting change of the gap height.

The stiffness s of the bushing can be calculated by numerically differentiating the load bearing capacity over the gap height.

$$s = - \frac{dF}{dh} \quad (4)$$

SETUP

In order to compare the properties of a bushing at ambient pressure and at vacuum conditions, the setup shall be operated within a vacuum chamber. The developed experimental setup is schematically shown in FIGURE 3 and pictured in FIGURE 4. The bushing is fixed within a housing and the shaft can be deflected by a drive mechanism. The actuation force is generated by a rotational drive outside of the chamber and feed through the chamber wall by a vacuum rotary feedthrough. The transmission of the rotational movement into the translational movement is accomplished with the help of a screw transmission and a lever. The induced force is measured by two load cells with a repeatability of 1N and the deflection is measured by capacitive displacement sensors with a resolution of 0.002 μm .

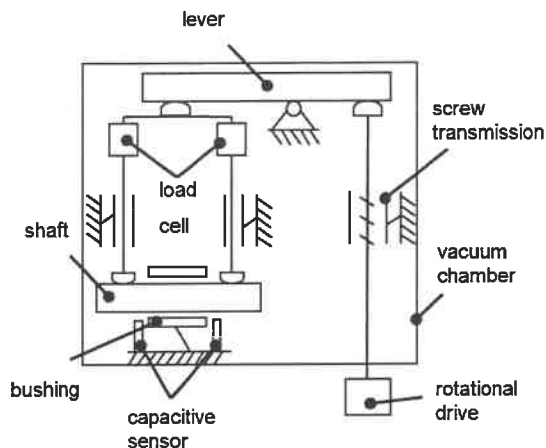


FIGURE 3. Schematic view of experimental setup for measuring load bearing capacity and stiffness of gas bushings

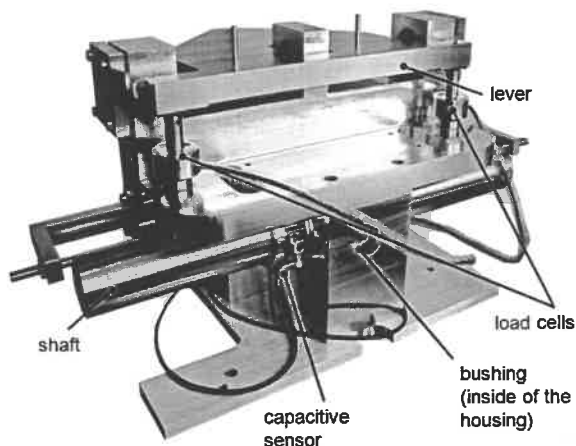


FIGURE 4. Picture of experimental setup for measuring load bearing capacity and stiffness of gas bushings

The pressure within the vacuum chamber equals the pressure of the first exhaust groove, enabling the experiments to be at normal atmosphere and at a low to medium vacuum in the range of 1 Pa to 50 Pa. For investigations at normal atmosphere, the vacuum chamber is opened up and for vacuum conditions, a rotary vane pump with a pumping speed of 0.01 m³/s is used for evacuating the vacuum chamber. The inlet pressure of the used nitrogen gas is measured using a pressure gauge with a accuracy of ± 350 Pa.

RESULTS

The minimal force acting on the shaft is the gravitational force of the shaft itself. To account for this force, the reference of every measurement is set without supply pressure to

the gap height $h = 0 \mu\text{m}$. With applied supply pressure, the measured gap height is assigned to the acting gravitational force of the setup of 40.1 N. Afterwards, the force is increased until a mean gap height of 1 μm is reached. The signals of the capacitive sensors and the load cells are recorded continuously during the measurement.

Three different absolute supply pressures ($p_s = 300,000$ Pa; $p_s = 410,000$ Pa and $p_s = 510,000$ Pa) were used. The surrounding pressure can be normal atmosphere (NA $\sim 100,000$ Pa) or vacuum (VA ~ 0 Pa). The load bearing capacity is measured at quasi-static conditions.

The mean load bearing capacity of ten measurements for selected mean gap heights of one bushing can be seen in FIGURE 5.

The mean load bearing capacity for a given gap height rises during the change from normal atmosphere to vacuum conditions. For the supply pressure of 300,000 Pa, the rise ranges from 9 % at the gap height of 13 μm to 21 % at 1 μm . For 410,000 Pa, the rise ranges from 9 % to 11 % and for 510,000 Pa the rise is almost constant at 7 %. Comparing the properties of the bushing for a constant relative supply pressure¹, the load bearing capacity at VA and $p_s = 410,000$ Pa compared to NA and $p_s = 510,000$ Pa decreases by 6 % to 8 %.

Compared to the data sheet of the manufacturer of the bushings, the stated load bearing capacity at NA and for a relative supply pressure of 410,000 Pa is above the measured results. The difference increases with decreasing gap height. The deviation ranges from 21 % at the gap height of 11.5 μm to 69 % at 1 μm gap height.

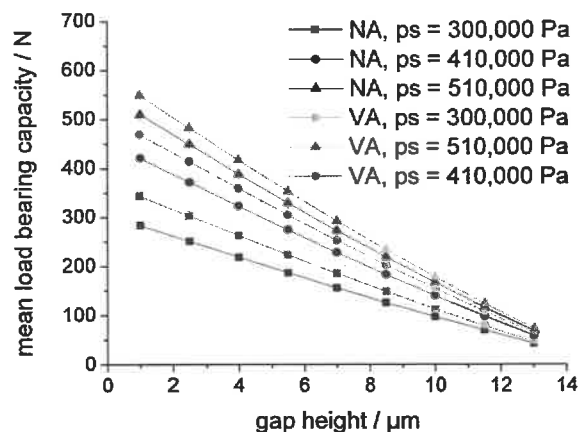


FIGURE 5. Mean load bearing capacity with different supply and ambient pressures over mean gap height

¹ difference between supply pressure and ambient pressure

measurements on each axis have been made with various feed rates in order to validate the model over the whole speed range of the axis. This is necessary since at different speeds and different torques the main source of heat losses changes.

In Fig 4 the comparison of the measured and simulated temperature values at the motor flange can be seen for three different feed rates of the X-axis, 500mm/min, 1500mm/min and 3000mm/min respectively.

Even though only parameters given by the manufacturer's documentation have been used, it can be seen that the correlation between the simulation and measurement is good. The method is desirable in order to predict the thermal behavior of machine tools, making subsequent parameter identification almost obsolete.

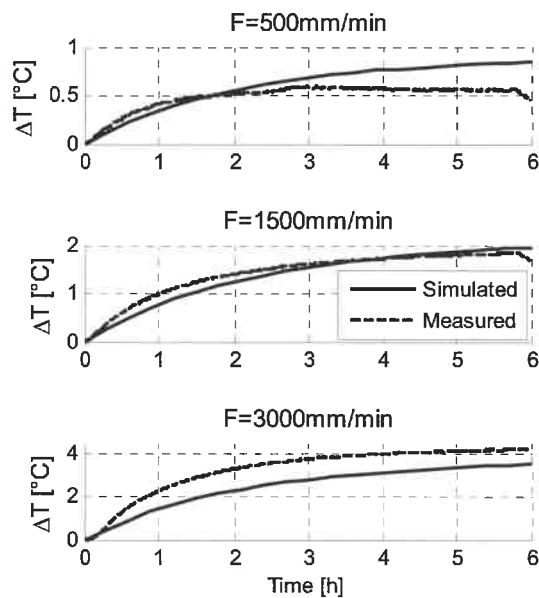


FIGURE 4. Comparison of measured and simulated motor flange temperatures for the X-axis at different feed rates.

The overall losses during the pendular movements are all rather small which is shown in Table 2. The losses increase with the feed rate mainly due to the increased loss in the bearing and also to a slight extent the iron losses. The required current, and therefore copper losses, are decreased with higher speeds because the torque curve of the whole drive train resembles the Stribeck curve. If higher feed rates would have been measured,

the torque, and as consequence the current, would increase again.

TABLE 2. Coefficients for different feed rates

Feed Rate [mm/min]	Loss [W]	Current [A]
500	1.30	0.46
1500	2.58	0.37
3000	4.75	0.33

CONCLUSIONS

It has been shown that using the model for the feed drive train, good agreements between the simulated and measurement temperature values at different feed rates can be achieved. Further validation using force measurements will be performed, in order to compare the simulated and measured load on the feed drive.

ACKNOWLEDGEMENT

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Static Stiffness Tuning Method of Linear Motion Guide of Machine Tool

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INSTRUCTION

A method has been developed to tune the static stiffness at a linear motion guide joint considering the whole machine tool system by interactive use of finite element method and experiment. This paper describes the procedure of this method and shows the results. For FEM simulation, the linear motion guide joint model is simplified using only guide, bearing and spring element. At the linear motion guide joint, the damping coefficient is ignored. The guide and bearing is connected by only spring element. By static experiment, 500 N is forced to the normal and lateral direction of bearing and the deformation is measured by capacitive sensor. The deformation by FEM simulation is extracted with changing the static stiffness of spring element. The tuned static stiffness is obtained by exploring the static stiffness directly trusting the deformation from the static experiment.

Background

There are many kinds of joints in machine tool. These machine tool joints are set up from various reasons such as transportation, function, design and manufacturing.

These joints have an important role in real processing. Particularly, the rotation joint and linear motion guide joint have very important considering that these joints have a role for coping principle.

The stiffness measurement and estimation of these machine tool joints are the most important factor on the design of machine tool considering that the machine tool design is based on the stiffness design. Up to now, the static stiffness of machine tool joints, such as spindle bearing, linear motion guide bearing and bolted joint, have been measured from the various kinds of the accurate methods. However, these methods did consider the static stiffness of only machine tool joint and the static stiffness considering whole machine tool have not been measured yet.

Therefore, for machine tool designer, it is necessary to measure the static stiffness of machine tool joint, which is considered from whole machine tool system.

In this paper, a tuning method that changes the static stiffness of only machine tool joint to the static stiffness considering the whole machine tool system. This method took the interactive use of the static experiment of whole machine tool system and finite element method of machine tool joints. And as the tuning target, the linear motion joint was selected.

Static Experiment

Figure 1 shows the appearance of static experiment of linear motion guide joint. This linear motion guide has four linear motion bearing(LM Block) and one ball screw system.

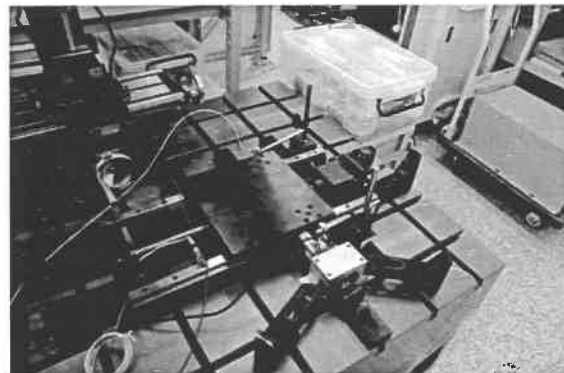


FIGURE 1. The static experiment of linear motion guide

The normal and lateral displacement of linear motion bearing was measured using capacitive sensors. At measuring the lateral displacement of linear motion bearing, the phenomenon that the linear motion guide system was pushed was considered and the displacement was the relative displacement which was measured using two sensors at the front and behind portion.

From these experiments, the static stiffness of linear motion block at the normal and lateral direction was obtained. Figure 3 shows the entire experimental system.

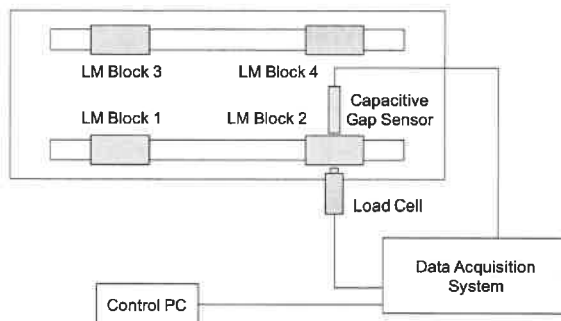


FIGURE 2. Experimental System

Static force 120 N is loaded at the normal direction five times and the normal displacement at each static force is averaged. And static force 2000 N is loaded at the lateral direction five times and the lateral displacement at each static force is averaged.

Table 1 and Table 2 show the averaged normal and lateral displacement at each LM block respectively.

Table 1. Normal displacement at each LM Block

	Normal displacement(μm)
LM Block 1	1.351
LM Block 2	1.476
LM Block 3	1.397
LM Block 4	1.483

Table 2. Lateral displacement at each LM Block

	Lateral displacement(μm)
LM Block 1	1.952
LM Block 2	2.174
LM Block 3	1.894
LM Block 4	2.224

Finite Element Method

With the assumption that the four linear motion bearing(LM Block) has the same normal and lateral stiffness, the whole linear motion guide system was modeled using 3D CAD software. Figure 3 shows the 3D CAD model of linear motion guide system. Using this 3D CAD model, the finite element method was conformed. In

finite element method, the static force was added at the linear motion bearing as same as the static experiment.

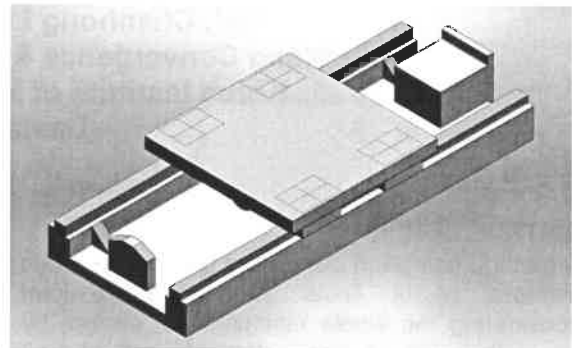


FIGURE 3. 3D CAD model of linear motion guide system

Figure 4 shows the mesh model of linear motion guide system and FEM tool Ansys 12.0 was used to analyze the displacement of linear motion guide system.

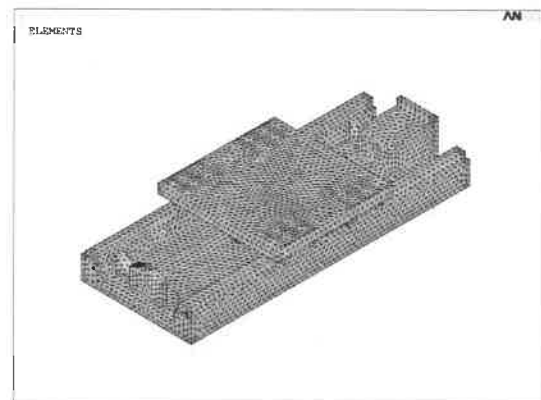


FIGURE 4. Mesh model of linear motion guide system

Above solid mesh model has the 60635 solid elements and 14885 nodes. As the structural materials, GC300 and SCM440 were selected. The structural properties of GC300 and SCM440 are shown at following table.

Table 3. Structural properties of GC300 and SCM440

	Density (kg/m^3)	Elasticity (GPa)	Poisson Ratio
GC300	7300	90.0	0.25
SCM440	7860	205.8	0.30

The stiffness element was located between guide way and LM block using *Matrix27* element like following figure 5.

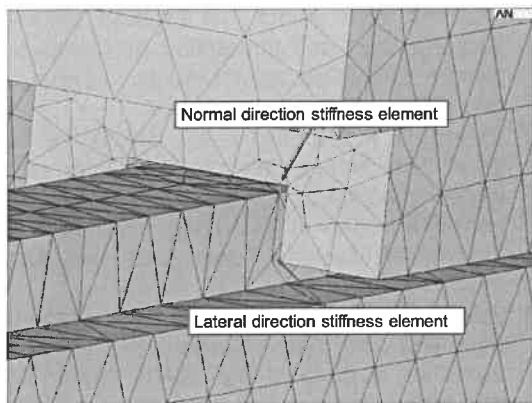


FIGURE 5. Locations of stiffness element of linear motion guide system between LM block and guide way

The ball screw was also considered as the stiff spring because of existence of its traveling direction. The stiffness value of ball screw is 278.87 N/mm according to catalog.

The initial value of normal stiffness was 240 N/ μ m and the initial value of lateral stiffness was 120 N/ μ m. The following figure 6 shows the displacement of linear motion guide system below the load normal static force 200 N. And the following figure 7 shows the displacement of linear motion guide system below the load normal static force 120 N. As the structural boundary condition, all degree of freedom at the bed of linear motion guide system was constrained.

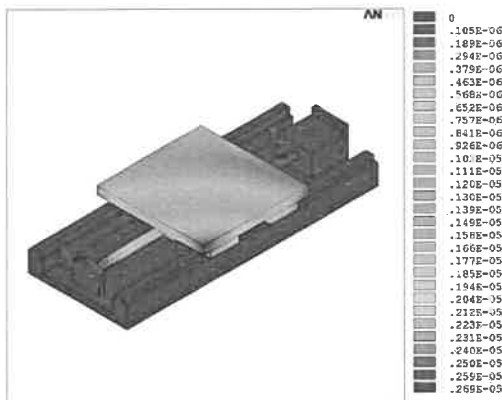


FIGURE 6. Displacement of linear motion guide system below the load normal static force 200 N

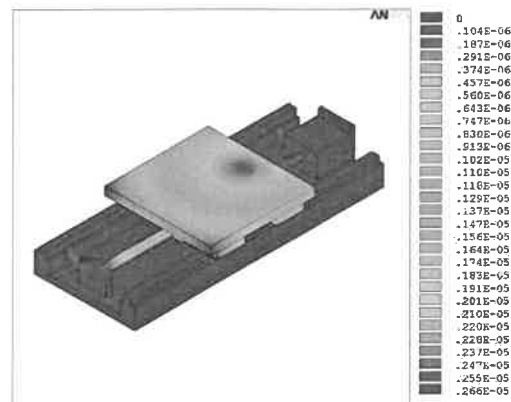


FIGURE 7. Displacement of linear motion guide system below the load lateral static force 120 N

Static stiffness tuning method

With the interactive use of experiment and finite element method, tuned static stiffness was obtained from regulation that the sum of difference of displacement from experiment and finite element method should be minimized.

This tuning method is the exploration method that with trusting the displacement from the static experiment the displacement from finite element method was explored.

The tuned stiffness was explored separating normal direction and lateral direction. In normal direction, the estimation function is given by following equation (1).

$$\Delta_s = \left(1 - \frac{\delta_{n1}}{1.351}\right)^2 + \left(1 - \frac{\delta_{n2}}{1.476}\right)^2 + \left(1 - \frac{\delta_{n3}}{1.397}\right)^2 + \left(1 - \frac{\delta_{n4}}{1.483}\right)^2 \dots\dots(1)$$

In lateral direction, the estimation function is given by following equation (2).

$$\Delta_s = \left(1 - \frac{\delta_{l1}}{1.952}\right)^2 + \left(1 - \frac{\delta_{l2}}{2.174}\right)^2 + \left(1 - \frac{\delta_{l3}}{1.894}\right)^2 + \left(1 - \frac{\delta_{l4}}{2.224}\right)^2 \dots\dots(2)$$

With above estimation functions, the normal and lateral stiffness were tuned at the direction that the sum of difference of displacement from experiment and finite element method should be minimized. And exploring was ended when the estimation function was smaller than 10^{-3} .

With varying the stiffness value of LM block, the displacements at each LM block were extracted. Substituting these displacement values to the estimation function, the value of estimation function was obtained.

Following figure 8 shows the tuning of normal stiffness and figure 9 shows the tuning of lateral stiffness.

	[N/ μ m]		[N/ μ m]
LMBlock 1 Stiffness	240		270
LMBlock 2 Stiffness	240		252
LMBlock 3 Stiffness	240		262
LMBlock 4 Stiffness	240		248
	[μ m]		[μ m]
LMBlock 1 Displacement	1.5197		1.3724
LMBlock 2 Displacement	1.5433		1.4762
LMBlock 3 Displacement	1.5195		1.4072
LMBlock 4 Displacement	1.5423		1.4952
Value of estimation function	0.026959737		0.000371914

FIGURE 8. Tuning of normal stiffness

	[N/ μ m]		[N/ μ m]
LMBlock 1 Stiffness	120		159
LMBlock 2 Stiffness	120		146
LMBlock 3 Stiffness	120		166
LMBlock 4 Stiffness	120		136
	[μ m]		[μ m]
LMBlock 1 Displacement	2.4890		1.9227
LMBlock 2 Displacement	2.5013		2.1461
LMBlock 3 Displacement	2.4883		1.9057
LMBlock 4 Displacement	2.4997		2.1742
Value of estimation function	0.212172901		0.000929572

FIGURE 9. Tuning of lateral stiffness

Discussion

Considering the normal stiffness of linear motion guide system, the tuned stiffness values of LM block were increased a little (LM block 1 and LM block 3 were increased about 10%, and LM block 2 and LM block 4 were increased about 15%). And the tuned lateral stiffness values of LM block were also increased a little (LM block 1 and LM block 3 were increased about 35%, and LM block 2 and LM block 4 were increased about 15%).

This is because of the existence of ball screw nut. Therefore other joints of machine tools have the effect on the stiffness of the stiffness of the other joint.

Conclusion

- The tuning method was proposed from the aspect that interactive use of experiment and finite element method.
- The tuning method was conformed using the LM block of linear motion guide system.
- The normal and lateral stiffness of LM block were affected by other machine tool joints, in this case, ball screw nut.

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DYNAMIC SURFACE ROUGHNESS PROFILER

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ABSTRACT

A dynamic profiler is presented that is capable of precision measurement of surface roughness in the presence of significant vibration or motion. Utilizing a special CCD camera incorporating a micro-polarizer array and a proprietary LED source, quantitative measurements were obtained with exposure times of $<100 \mu\text{sec}$. The polarization-based interferometer utilizes an adjustable input polarization state to optimize fringe contrast and signal to noise for measurement of optical surfaces ranging in reflectivity from 1 to 100%. In addition to its vibration immunity, the system's light weight, $<5 \text{ kg}$, compact envelope, $24 \times 24 \times 8 \text{ cm}$, integrated alignment system, and multiple mounting options facilitate use both directly resting on large optical surfaces and directly mounted to polishing equipment, stands, gantries and robots. Measurement results presented show an RMS repeatability $<0.005 \text{ nm}$ and an RMS precision $<0.1 \text{ nm}$ which are achieved without active vibration isolation.

INTRODUCTION

A significant challenge to high precision on-machine metrology is the requirement for vibration control and or significant vibration immunity. For interferometric measurements, the primary technique for obtaining vibration immunity is rapid measurement acquisition, typically sub 100 usec. One of the principal techniques for rapid phase measurement is the spatial carrier phase shifting method which utilizes a single interferogram to extract phase information. [1] The primary advantage of the spatial phase measurement technique over temporal phase measurement is that only one image is required, allowing acquisition times several orders of magnitude smaller than in temporal phase shifting. Rapid acquisition offers both significant vibration immunity, and the ability to measure dynamic events.

A novel approach to spatial phase measurement has been developed by 4D Technology Corporation called the *pixelated mask spatial*

carrier method. [2-4] In this technique, the relative phase between the carrier and the test wavefront is modified on a pixel-by-pixel basis by a micro-polarizer phase shifting array placed just prior to detection. This technique has been successfully implemented in commercial vibration insensitive Twyman-Green and Fizeau interferometers with wavelengths from the infrared through UV. [5-8]

This paper presents a portable, high precision, vibration immune surface roughness profiler which utilizes 4D Technology's pixelated phase mask sensor. This paper is arranged in the following manner. In the first section the theory of operation is presented and design challenges discussed. Next, the prototype system is presented along with several different mounting schemes. Section 4 discusses the theory and performance of a new phase calculation algorithm. Finally, in section 5 performance measurement results are presented and discussed.

THEORY OF OPERATION

The basic layout of the dynamic surface roughness profiler is shown in figure 1. Functionally, the dynamic profiler is an epi-illuminated polarization interference microscope utilizing a Linnik interference objective and 4D Technology's pixelated polarization mask based phase sensor.

Functional Layout

This system is composed of three major sections: (1) A Kohler illuminator utilizing a high brightness 460nm LED source with adjustable output polarization. (2) A Linnik interference objective with orthogonally polarized test and reference channels, and (3) A 1.5 Mpix, low noise CCD camera with a pixelated polarization mask.

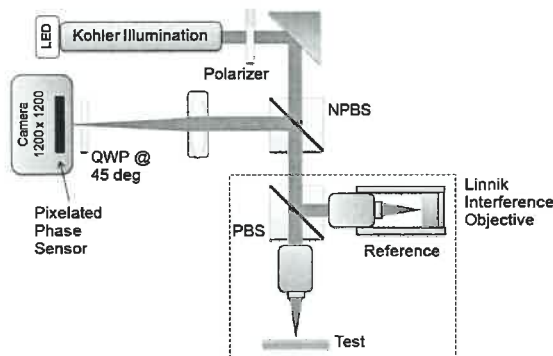


FIGURE 1. Functional layout of the dynamic surface roughness profiler.

Operational Description

System illumination is provided by the Kohler illuminator with an adjustable polarizer at its output. The illumination path is folded into the microscope imaging path via a non-polarizing 50/50 beam splitter. Per definition, the Kohler illuminator images the source at the entrance pupil of the Linnik objectives.

The polarization beam splitter in the Linnik objective sends the p -polarized light to the test object and the s -polarized light to the reference mirror. The reference mirror is a coated fused silica super smooth flat with a 25% reflectivity. The test to reference intensity beam ratio may be adjusted with the illumination system polarizer, allowing optimization of fringe contrast.

Upon return from the test and reference surfaces, the test and reference beams recombine at the polarizing beam splitter and exit the Linnik objective. Both beams then pass back to the non-polarizing beam splitter where half of the intensity is directed toward the camera via the system tube lens which images the object and reference surfaces at the CCD camera.

Just prior to the camera both beams pass through a quarter wave plate at 45 degrees which converts the p -polarized test beam into right-hand circular polarization and the s -polarized reference beam into left hand circular polarization. Both beams then pass through the pixelated polarization mask forming an interference pattern at the detector.

Design Challenges

There are two significant challenges associated with use of the pixelated phase sensor in a microscope configuration. The first challenge is obtaining a compact source with the necessary brightness to allow short integration times. The second is polarization control necessitated by the requirement that the test and reference beams be orthogonally polarized for the pixelated polarization mask phase sensor.

Illumination Source

The primary factors driving source selection were integration time, size, and heat load. In order to maintain integration times below 100usecs when measuring objects with 1% reflectivity, a high brightness source was necessary. This requirement coupled with the desire for a small form factor and minimal heating led to the development of a proprietary 460nm LED source. The source is used in a Kohler illumination arrangement with a polarizer at the output and is fed into the imaging path through a non-polarizing beam splitter. Adjustment of the illumination polarization allows maximization of fringe contrast for different test object reflectivities.

Polarization Control

Use of the pixelated phase sensor requires that the test and reference beams of the interferometer be orthogonally polarized. Due to the narrow angular tolerance on polarization control coatings it is necessary to place the polarization beam splitter used in the interference objective in collimated or nearly collimated space. Of the three standard microscopic interference objectives, Michelson, Mirau, and Linnik, only the Linnik objective layout allows placement of the polarizing beam splitter in the pseudo-collimated space at the objective entrance pupil. Because of this placement, the Linnik objective type offers the largest working distance of all interference objectives which provides a margin of safety especially for magnifications greater than 50X. The polarization requirement coupled with the desire to have a compact design presented further challenges due to the number of optical folds in the beam path.

PROTOTYPE DESIGN

For maximum flexibility, a compact and lightweight profiler head is a requirement. The prototype system designed, figure 2, weighs less than 5 kg and fits into a compact envelope of 24 x 24 x 8 cm. The compact and lightweight design enables the head to be lifted with one hand for repositioning on a measurement surface. Its light weight also simplifies the requirements and cost for positioning systems such as gantries or robots.

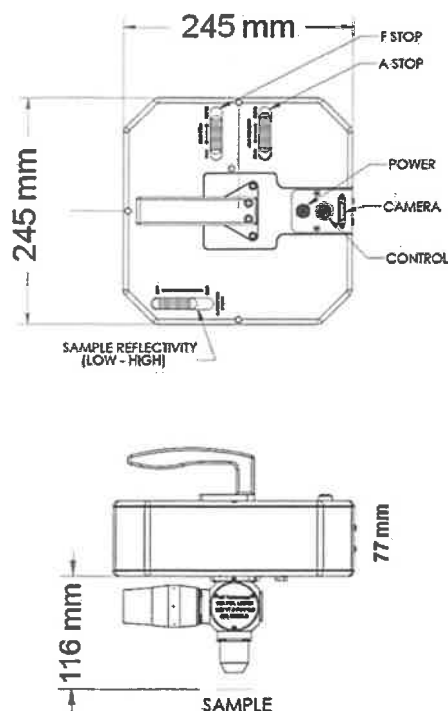
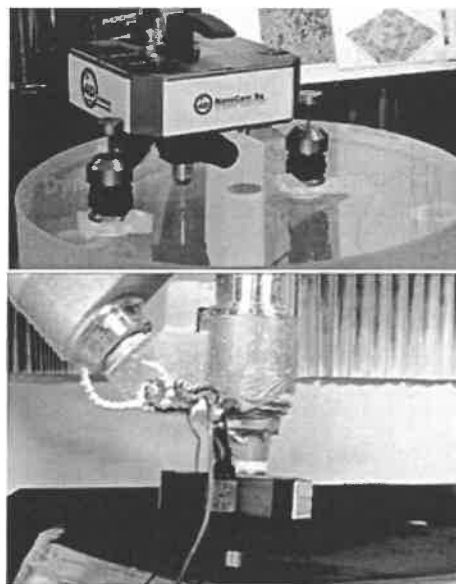


FIGURE 2. Schematic diagram of the dynamic surface roughness profiler head.

Mounting Schemes

A versatile and flexible mounting scheme was designed to meet the very different mounting requirements for large optic manufacturers. The primary means for mounting the profiler is a tilt and focus adjustable tripod that enables the profiler to be mounted directly onto the surface of a large optic. Lens tissue can be secured to the Teflon tripod contact pads to protect the surface. Small optics can be placed directly under the tripod-mounted profiler making it capable of measuring an optic of any diameter. The profiler can also be mounted directly to a polishing robot with a chuck interface located on the top of the

head or to existing gantries or stands with a dove-tail interface located on backside. Figure 3 shows the engineering prototype of the dynamic surface roughness profiler mounted to a tripod base, top, and mounted to a polishing machine, bottom.



Courtesy of Zeeko Ltd

FIGURE 3. Dynamic surface roughness profiler mounted to tripod, top, and robotic polishing tool on right.

MEASUREMENT PERFORMANCE

General performance specifications for the dynamic surface profiler are listed in Table 1. These specifications were determined through measurements of a super smooth surface on a non-vibration isolated table.

TABLE 1. Dynamic surface profiler general measurement specifications.

	RMS (nm)	PVq (nm)
Shot Noise	0.55	
Uncalibrated Accuracy		
Low Order (shape)	0.98	3.0
High Order (roughness)	0.1	0.4
16 Averages		
Precision	0.08	0.3
Repeatability	0.004	0.01

One of the advantages of the dynamic surface roughness profiler over temporal or vertical scanning systems is the ability to average a very large number of measurements. One technique for determining measurement noise is by taking

difference measurements. As long as the system is stable, the difference measurement should be uniformly distributed shot noise with an RMS value equal to $\sqrt{2}$ times the single measurement value. Figure 4 shows the measured vs. theoretical shot noise values for 16 through 2000 average difference measurements taken on a super smooth flat with a 10X objective. The measurements were taken on a non vibration isolated table. These measurements demonstrate the ability of the dynamic surface profiler to obtain very large average measurements with shot noise limited performance over time scales exceeding 1 minute.

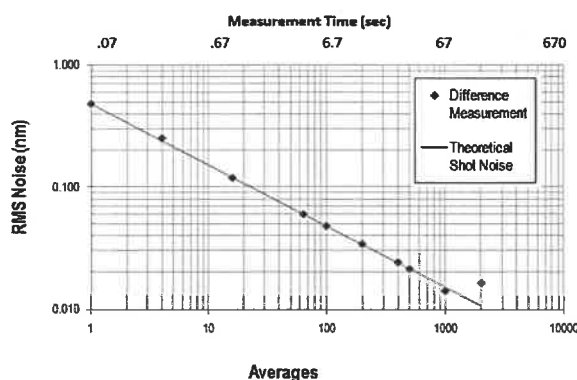


FIGURE 4. Measurement noise of the dynamic surface profiler.

The effect of averaging on RMS precision and repeatability were measured on a super smooth flat with the 10X objective. The results of these measurements are shown in figure 5. Both precision and repeatability trend as expected with an increasing number of averages. It is not until >1000 averages that the trend in performance benefit with averaging starts to decrease.

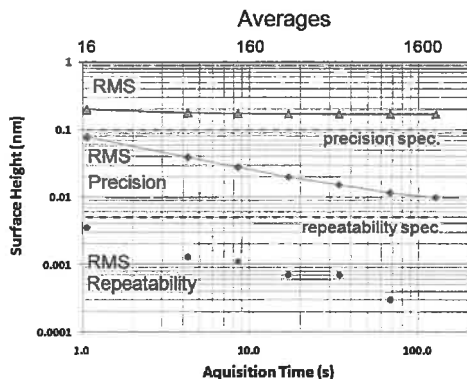


FIGURE 5. Surface profiler RMS precision and repeatability measurements as a function of measurement averages.

SUMMARY

We have developed a dynamic surface profiler using a polarization based interference objective and pixelated micro-polarizer camera that is capable of measuring surface roughness in the presence of significant vibration. To achieve shot noise limited performance, we developed a correction algorithm that eliminates phase-dependent measurement error. We demonstrated the ability to measure with sub-angstrom repeatability and precision through modest averaging. Measurements with 5x, 10x, and 20x objectives show Nyquist limited performance. The portability, high measurement quality, and short acquisition time of the prototype system allow its use in a wide variety of applications ranging from surface roughness in large optics manufacturing to capture of dynamic events in MEMS actuators.

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MACROSCOPIC TRANSFER ARTIFACT FOR NANO-POSITIONING

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INTRODUCTION

The need for instruments that produce nano-structures with higher throughput and measure at the sub-nanometer level led to the development of the MITs Nano-ruler [1] and the Sub-Atomic Measuring Machine (SAMM) [2,3]. The motivation of the Nano-ruler was to build extremely precise gratings called "Nano-grids" and in one application they will be utilized as optical encoder plates to detect work piece motion with 0.1 nm precision. This economical technique with improved dimensional accuracy will have an influence on advancing the metrology in future nano-lithography tools [1]. In order to quantify and reduce errors in the Nano-ruler to achieve 0.1 nm pattern measurement repeatability it must be thoroughly tested using another instrument. The Sub-Atomic Measuring Machine (SAMM) at UNC Charlotte will be utilized to measure gratings manufactured by the Nano-ruler and also compare the reference frames of both Nano-grids and the SAMM interferometer.

A highly accurate atomic force microscope has been developed at MIT and is being integrated with SAMM stage that can position and measure with a resolution of approximately 0.01 nanometers [3]. The systematic errors of SAMM must be removed to reach the target uncertainty. Since SAMM is one of the leading machines of its kind there are no known calibrated artifacts that exceed SAMM's resolution. A self-calibration technique [4] with a newly developed artifact will be utilized to error map the SAMM.

The artifact will also be used as a transfer standard to measure internationally on other machines with similar capabilities to determine the uncertainty of these instruments. The artifact includes periodic structures nominally equally spaced in a grid pattern that should remain dimensionally stable over time. The features must be able to be identified as points. Lines are included between the features to assist in finding the structures. In this paper,

these features will also be called points, inverted pyramids, or pits. The algorithm used to determine the systematic errors will follow Ye's [5], therefore, the artifact must be able to be translated the same distance that the features are spaced from each other and rotated 90°. The artifact must be able to withstand cleaning procedures prior to measurement without causing damage to the features or artifact itself.

It was decided to fabricate the artifact from single crystal Silicon (Si) due to its presumed temporal stability, cost, ability to etch along the crystalline planes, and the extensive published knowledge on various manufacturing and cleaning processes.

This paper describes the design of the transfer artifact, provides a brief summary of the final fabrication process and shows AFM images of an inverted pyramid on the artifact.

BACKGROUND

Preferential Etching

Preferential etching is also called structural etching and offers a higher degree of selectivity between the different crystalline planes. The geometry and size of the pit is dependent on the mask features and wafer orientation.

Predictable inverted pyramids or v-grooves are easily fabricated when the appropriate features on the mask are accurately aligned to bound the (111) plane. The primary orientation flat on the wafer is roughly aligned to this direction. The {111} planes that have the largest packing density in contrast to the other planes and are fundamentally not attacked during the etch, which create the structures as illustrated in Figure 1 [6].

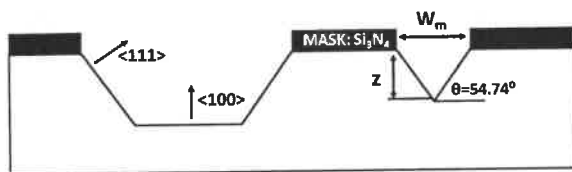


FIGURE 1. Diagram of structures produced using preferential etching and (100) oriented Si wafer

This figure demonstrates the structures formed using a square or rectangular mask opening (W_m) and a Silicon Nitride (Si_3N_4) hard mask. If the wafer is etched deep enough in relation to W_m , it will intersect the (100) bottom plane parallel to the surface and form a pit. An angle of 54.74° may be achieved by etching (100) oriented wafers. If W_m is the mask opening width, W_0 is the bottom cavity width, and z is the etch depth, W_0 may be calculated using the equation below [6].

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If it is desired to construct a pit, $W_0=0$, and W_m is

Although anisotropic wet etching of Si is advantageous by accomplishing control in the lateral direction, the etch rates are slower in the <100> direction compared to isotropic etchants. Two crystal dependent etchants are potassium hydroxide (KOH) and tetramethyl ammonium hydroxide (TMAH).

CURRENT DESIGN

As the thickness of a plate increases the bending due to gravity decreases. The maximum cosine error of the point placement of the standard plate was computed using both Loewen's optimal support points (best case scenario) [7] and modeling the plate as a cantilever beam using Roark's equation (worst case scenario) [8]. The maximum cosine errors modeling as a plate using Loewen's optimal points and modeling as a cantilever beam were 0.00075 and 3.05013 picometers, respectively. A thickness of 1 mm was chosen because the displacement error was well below the resolution of SAMM and that is the maximum thickness allowed for most of the equipment used to fabricate the artifact in the clean rooms at UNC Charlotte.

It was desired to obtain a Si substrate that is ideally flat and "stress-free" material. To prevent any stresses due to crystal dislocations in the

material resulting from impurities in the Si, intrinsic material is desired. Typically, float zone wafers are used for devices that require low dopant levels because there are less oxygen impurities caused from contact to the silica crucible used in the Czochralski (Cz) growth process [6]. When purchasing wafers many factors were considered such as cost, resistivity (dopant concentration), thickness, polishing process, total thickness variation and the alignment of the flat with the crystalline plane. It was difficult to find cost effective float zone wafers with such requirements. After an extensive search, it was decided to purchase (100), p-type, boron doped, prime Cz Si wafers from James River Semiconductor [9]. The substrates were 1 mm thick, with a total thickness variation of $2\text{ }\mu\text{m}$, double side polished, 1-10 ohm-cm resistivity and the flat was 0.5° aligned with the crystalline plane.

The front side of the artifact includes 0.004 mm inverted pyramids etched into the [100] plane with a pitch of 1 mm and narrow grooves in between allowing easy location of the measurement points. Figure 2 displays a SolidWorks model of the grid plate design.

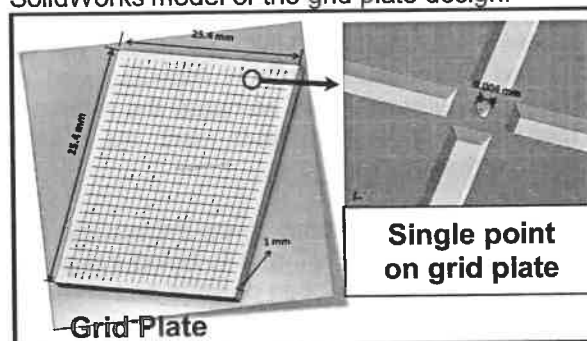


FIGURE 2. SolidWorks model of the front side of Artifact

Originally a three v-groove kinematic mount was designed for the back of the artifact to allow for quick placement while measuring in the three positions. Spheres would have been attached to an intermediate plate. Since it is necessary to translate the artifact by 1 mm, the diameter of the balls was planned to be less than 1 mm. It was not possible to preferentially etch v-grooves for the mount because preferentially etched grooves need to be aligned with the crystal planes so grooves were made on a separate wafer cut into 5mm x 6mm pieces and would have been fusion bonded onto the artifact. Figure 3 illustrates the back side of the artifact after the proposed bonding.

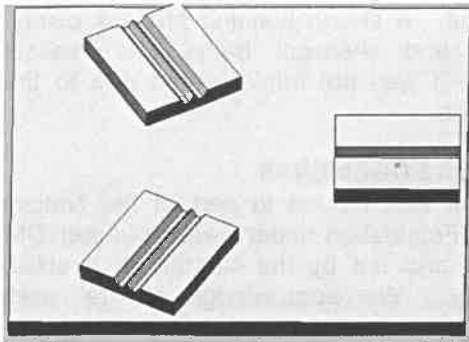


FIGURE 3. Back side of artifact

There were two sets of balls to be glued on the intermediate plate rotated 90° from each other. The optimal points were chosen using Loewen's equations [7]. Figure 4 illustrates the planned modes of operation of the artifact.

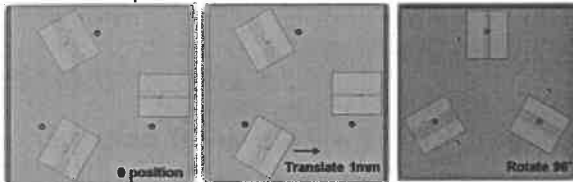


FIGURE 4. Operational modes of artifact.

The artifact will be measured in zero position. Then the artifact would have been translated by 1 mm on the second set of v-grooves and measured. The last position of the artifact is the rotation by 90° on the second set of balls. Since the grooves were not implemented alignment will be done on an optical measuring machine (Nikon MM-400) with the artifact on a kinematically supported sub-plate.

FABRICATION PROCESS

A hard mask was used instead of photo-resist in the lithography process. The wafers were coated with a 250nm of Si_3N_4 on both sides using Plasma Enhanced Chemical Vapor Deposition (PECVD) to serve as a hard mask. A 1x contact photolithography mask was manufactured by Photosciences [10]. It included both the features to pattern the grid plate design and the v-grooves for the kinematic mount.

The Si_3N_4 was etched using an Reactive Ion Etch (RIE) system. The v-grooves were then separated from the grid plate using a Diamond Saw. The organics on the samples were removed by immersing in a piranha bath. Prior to etching the silicon, the samples were placed in a Buffered Oxide Etch (BOE) mixture to remove the native oxide.

Preferentially etching was performed using a stirrer hotplate with a temperature probe. The substrates were placed in a stirred bath at an angle. The grid plate was etched using TMAH (25% concentrated) and 10% isopropyl alcohol (IPA) at 90°C for 6minutes. The deep v-grooves for the kinematic mount were etched using KOH and IPA at 60C for 25hours.

It was important to develop correct etch rates of both the Si_3N_4 and the Si during both etches. If either were over etched, it caused severe undercutting as shown in Figure 5.

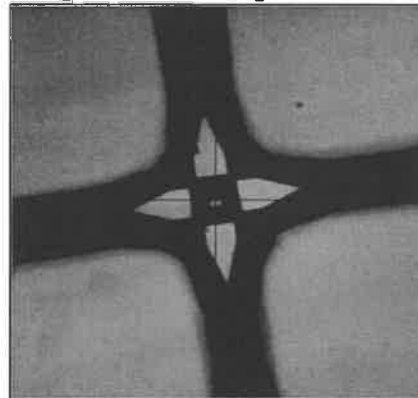


FIGURE 5. Undercutting of the mask of grid plate design.

The mask undercut caused the inverted pyramids and the narrow grooves to intertwine.

The artifact was bonded with wax to a carrier wafer and cut into a 1 in by 1 in square using a Diamond Saw. The artifact was then cleaned with piranha and the Si_3N_4 was stripped using BOE.

A fusion bonding process of silicon was developed, however, due to time constraints the v-grooves were not bonded to the artifact.

INSPECTION

All 625 squares of the artifact were inspected using a microscope to assure all features had etched. An atomic force microscope was used to evaluate the inverted pyramids on the artifact. The tip of the AFM was not calibrated, therefore, there could be some aliasing of the measurement points. Figures 6 and 7 display the AFM images of one of the points on the artifact.

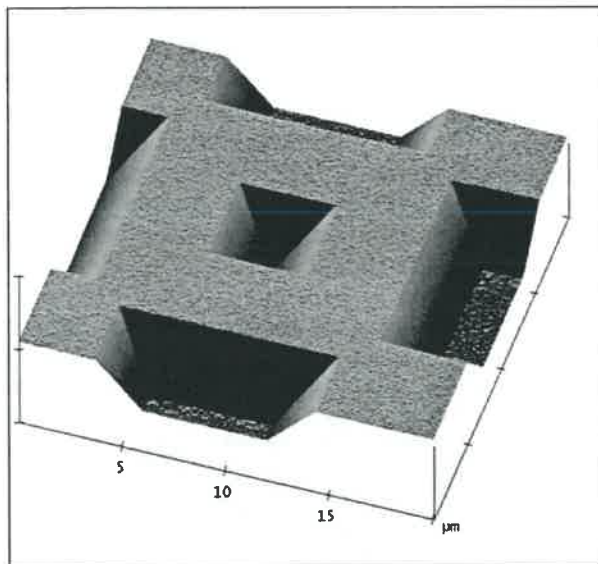


FIGURE 6. AFM Image of the inverted pyramid and the surrounding narrow grooves.

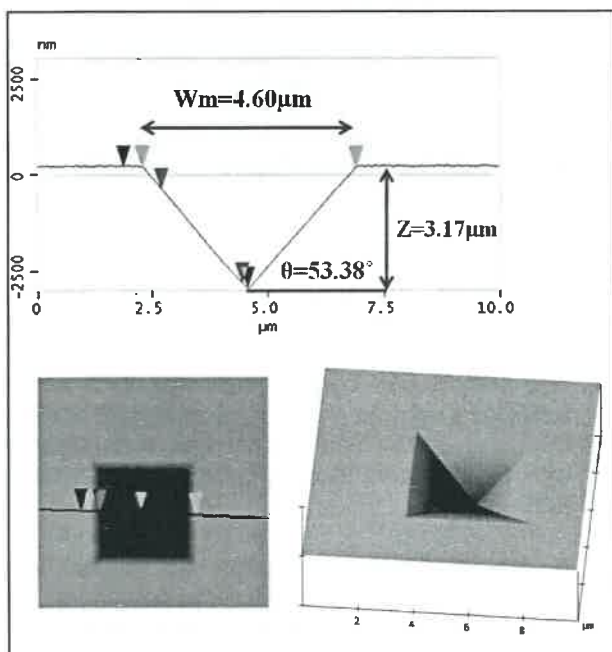


FIGURE 7. AFM image of the inverted pyramid.

The width opening of the artifact, the depth, and the angle of the sidewall were $4.6\mu\text{m}$, $3.17\mu\text{m}$, and 53.38° , respectively.

CONCLUSION

The artifact that will be used to error map the SAMM stage has been manufactured. It will also be measured on other machines and all results will be compared.

The entire fabrication process of the artifact was developed. A silicon bonding process using a hotplate and chemical baths was realized, however, it was not implemented due to time restrictions.

ACKNOWLEDGEMENTS

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MODELING OF BALLSCREW NUT STIFFNESS INCLUDING THERMAL EFFECT

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ABSTRACT

A nut stiffness model including thermal effect is proposed to develop an accurate stiffness model of a ballscrew system. Differential temperature distribution is estimated through the finite difference method (FDM). The differential growth is applied for obtaining the stiffness according to assembly and load conditions..

INTRODUCTION

Ballscrew are important motion transfer and positioning elements of industrial machinery and precision machines. Positioning accuracy of the system depends upon geometric error and axial stiffness of feeddrive systems. To compensate for the positioning error, and to predict accuracy of precision machines during the design process, accurate stiffness estimation of the feeddrive system equipped with ballscrews is required [1]. With the development of high-speed machining and motion devices in the electronics industry, thermal characteristics of a precision machine have been held of much attention in the industry. Thermal error generated from the high-speed motion is critical to confirm the positioning performance of the precision machinery.

Bossmanns and Tu [2] developed a model to analyze the heat generation, heat transfer and heat sink of a high-speed spindle. Otsuka and Fukuda [3] studied thermal expansion of ballscrews through the identification of heat generation mechanism between the screw and nut. Chung and Park [4] proposed a finite difference method (FDM) for thermal analysis of ballscrews.

Ballscrew expansion due to friction and heat degrades accuracy of precision machines. To calibrate the thermal deformation, an accurate estimation of the whole temperature field of the screw is required first. However, it is inefficient and impossible to acquire the whole temperature

distribution by the temperature measurement of every point. So, a thermal model, which estimates the temperature from a mathematical model, is required. In the previous study [5], the one-dimensional heat transfer of a ballscrew was formulated through the eigenvalue and state-space analyses. And then an observer was designed to estimate the intensity of heat source and the whole temperature field in real time by solving the inverse heat problem. Thermal expansion of a ballscrew was calibrated from the estimated temperature distribution. However, thermal contact resistance at contact points between balls and grooves, and the change of heat generation of the ballscrew mechanism have been ignored. When balls and grooves of the ballscrew are in contact, the presence of deformation and friction generates heat. Then the nut stiffness varies with the preload change. These effects change the previously developed ballscrew nut stiffness model [1].

In this paper, FDM is applied to analyze the temperature distribution and thermal expansion of a ballscrew system. Estimation of the preload change owing to temperature differentials between the nut and shaft elements is conducted. Thermal nut stiffness model of a ballscrew is proposed according to assembly, loading and operating conditions.

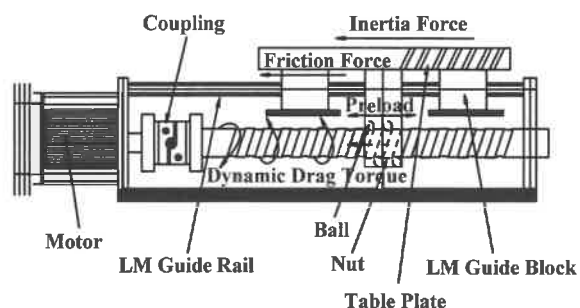


FIGURE 1. Double-nut Ballscrew Feeddrive System.

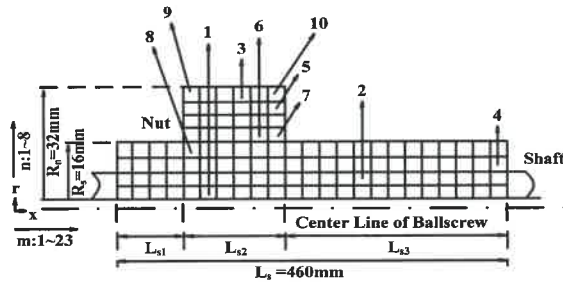


FIGURE 2. Axisymmetric FDM Model of the Ballscrew Structure.

THERMAL MODELING

Considering heat generation q' due to the drag torque, heat conduction equation of the ballscrew shown in Fig. 1 is given by

$$\frac{\partial^2 T}{\partial x^2} + \frac{1}{r} \frac{\partial T}{\partial r} + \frac{\partial^2 T}{\partial r^2} + \frac{q'}{k} = \frac{1}{\alpha} \frac{\partial T}{\partial t} \quad (1)$$

where $\alpha = k/\rho c$ is thermal diffusivity and k is thermal conductivity. Initial condition is $T = T_\infty$. For the axisymmetric FDM model shown in Fig. 2, boundary conditions are given as follows:

$$\left. \begin{aligned} x=0, \quad 0 < r < R_s, \quad T &= T_B \\ x=L_s, \quad 0 < r < R_s, \quad T &= T_B \\ r=0, \quad 0 < x < L_s, \quad \frac{\partial T}{\partial r} &= 0 \\ r=R_s, \quad L_{s1} < x < L_{s2}, \quad T_s &= T_{fs} : T_n = T_{fn} \\ r=R_s, \quad 0 < x < L_{s1} \\ r=R_s, \quad L_{s1} + L_{s2} < x < L_{s3} \\ x=L_{s1}, \quad R_s < r < R_n \\ x=L_{s2}, \quad R_s < r < R_n \\ r=R_n, \quad L_{s1} < x < L_{s2} \end{aligned} \right\}, \quad -k \frac{\partial T}{\partial r} = h(T - T_\infty) \quad (2)$$

where h is heat convection coefficient around the surfaces, T_B is supported bearing temperature, and T_{fs}, T_{fn} are contact temperatures of the shaft and nut, respectively. L_{s2} is nut length, L_{s1}, L_{s3} defines the stroke range of the nut.

Frictional Heat Generation

Dynamic drag torque caused by thrust force and preload generates frictional heat between balls and grooves. The heat diffuses to contact elements of the shaft and nut. The frictional heat per unit element and time is given by [3,4]

$$Q_f = \frac{\pi N T_d}{60 I} \quad (3)$$

where N denotes rotational speed (rpm) of the shaft, T_d means drag torque (N·mm) and I is the number of elements of the nut in the axial direction. Considering volume elements of the shaft and nut, and the heat equilibrium of the elements, boundary temperatures of the shaft and nut during Δt due to the drag torque at the contact elements are given by [3,4]

$$T_{fs} = \frac{Q_f \Delta t}{c \rho V_s}, \quad T_{fn} = \frac{Q_f \Delta t}{c \rho V_n} \quad (4)$$

TABLE 1. Parameters for the FDM Analysis.

Thermal Diffusivity α	$1.67 \times 10^{-5} \text{ m}^2/\text{s}$
Specific Heat c	$460 \text{ J}/(\text{m}^3 \text{ } ^\circ\text{C})$
Density ρ	$7865 \text{ kg}/\text{m}^3$
Heat Conductivity k	$60.5 \text{ W}/(\text{m}^\circ\text{C})$
Thermal Expansion Coefficient σ	$11.7 \times 10^{-6} 1/^\circ\text{C}$
Energy flow Ratio between Nut and Shaft R	$0 \sim 0.5$
Ambient Temperature T_∞	0°C
Shaft Radius R_s	32 mm
Outside Nut Radius R_n	64 mm
Nut Length L_{s2}	120 mm
Shaft Length L_s	460 mm
Ball Diameter	3.175 mm
Shaft Revolution Speed	500, 1000, 1500 rpm
Preload	100 kgf
Axial Difference Δx	20 mm
Radial Difference Δr	4 mm
Time Difference	0.03 sec

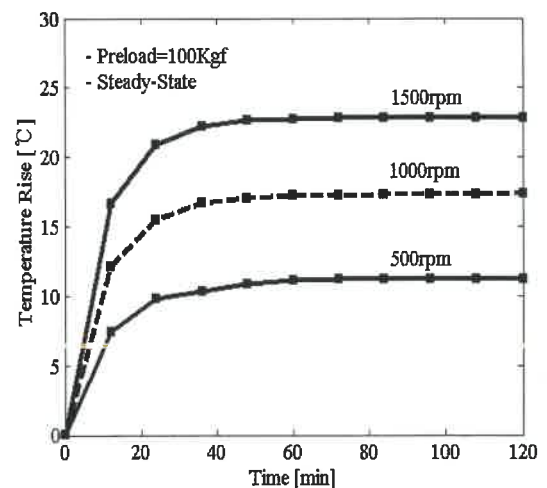


FIGURE 3. Temperatures vs. Operating Time of the Shaft Surface Element at $x=140\text{mm}$.

Finite Difference Method (FDM)

Fig. 2 shows an axisymmetric finite difference model with the axial and radial differences Δx and Δr , respectively. This finite difference model consists of coaxial cylindrical pipe elements. They are axially and radially stacked together. A total of 116 elements are defined as shown in Fig 2. Using the heat equilibrium condition, heat conduction in the solids, heat convection around the surfaces and the previous friction heat generation mechanism, explicit form of finite difference equations for the (m,n) -th element is obtained according to the heat transfer types (1 ~ 10 of Fig. 2) as given by Eq. (5).

$$1: T_{m,1}' = T_{m,1} + Fox(T_{m+1,1} + T_{m-1,1} - 2T_{m,1}) + 4For(T_{m,2} - T_{m,1})$$

$$2: T_{m,n}' = T_{m,n} + Fox(T_{m+1,n} + T_{m-1,n} - 2T_{m,n}) + For(k_1 T_{m,n-1} + k_2 T_{m,n+1} - 2T_{m,n})$$

$$3: T_{m,n}' = T_{m,n} + Fox(T_{m+1,n} - T_{m-1,n} - 2T_{m,n}) + For \cdot k_3(T_{m,n-1} - T_{m,n}) + For \cdot Bir \cdot k_4(T_{\infty} - T_{m,n})$$

$$4: T_{m,n}' = T_{m,n} = T_B$$

$$5: T_{m,n}' = T_{m,n} + For(k_1 T_{m,n-1} + k_2 T_{m,n+1} - 2T_{m,n}) + Fox(T_{m-1,n} - T_{m,n}) + Fox \cdot Bir(T_{\infty} - T_{m,n})$$

$$6: T_{m,n}' = T_{m,n} + Fox(T_{m+1,n} + T_{m-1,n} - 2T_{m,n}) + For \cdot k_5(T_{m,n+1} - T_{m,n}) + R \frac{T_{m,n-1} V_s - T_{m,n} V_n}{V_n} + T_{f,n}$$

$$7: T_{m,n}' = T_{m,n} + Fox(T_{m-1,n} - T_{m,n}) + For \cdot k_5(T_{m,n+1} - T_{m,n}) + Fox \cdot Bir(T_{\infty} - T_{m,n}) + R \frac{T_{m,n-1} V_s - T_{m,n} V_n}{V_n} + T_{f,n}$$

$$8: T_{m,n}' = T_{m,n} + Fox(T_{m+1,n} - T_{m-1,n} - 2T_{m,n}) + For \cdot k_5(T_{m,n-1} - T_{m,n}) + R \frac{T_{m,n+1} V_n - T_{m,n} V_s}{V_s} + T_{f,s}$$

$$9: T_{m,n}' = T_{m,n} + Fox(T_{m+1,n} - T_{m,n}) + For \cdot k_3(T_{m,n-1} - T_{m,n}) + (2Fox \cdot Bir + k_4 For \cdot Bir)(T_{\infty} - T_{m,n})$$

$$10: T_{m,n}' = T_{m,n} + Fox(T_{m-1,n} - T_{m,n}) + For \cdot k_3(T_{m,n-1} - T_{m,n}) + (2Fox \cdot Bir + k_4 For \cdot Bir)(T_{\infty} - T_{m,n})$$

where

$$k_1 = 1 - \frac{0.5}{n}, k_2 = 1 + \frac{0.5}{n}, k_3 = \frac{4(2n-1)}{4n-1}, k_4 = \frac{8n}{4n-1}, k_5 = \frac{4(2n+1)}{4n+1}$$

$$Bir = \frac{h\Delta x}{k}, Bir = \frac{h\Delta r}{k}, Fox = \frac{\alpha\Delta x}{(\Delta x)^2}, For = \frac{\alpha\Delta r}{(\Delta r)^2}$$

$$V_s = \frac{\pi(n-0.25)(\Delta r)^2 \Delta x}{2}, V_n = \frac{\pi(n+0.25)(\Delta r)^2 \Delta x}{2}$$

(5)

In Eq. (5), T' means next step temperature and T is the current. The drag torque and contact temperatures given by Eq. (3) and (4) are applied for elements between 6 and 8 type nodes along the axial direction. Using the parameters given in Table 1, Eq. (5) is solved with boundary conditions given in Eqs. (3) and (4). Fig. 3 shows temperatures of the shaft surface element at $x=140$ mm node position according to operating time of the ballscrew. Results show that the warm-up time is around 2 hours. As frictional heat produces differential thermal growth that increases preload of the ballscrew, steady-state temperatures increase according to the rotational speed. Fig. 4 shows the whole temperature distribution with respect to the nut location, operating speed and preload. As the ambient temperature is selected 0°C , displayed ones are differential temperatures.

Preload Change

At steady-state, temperature distributions at an

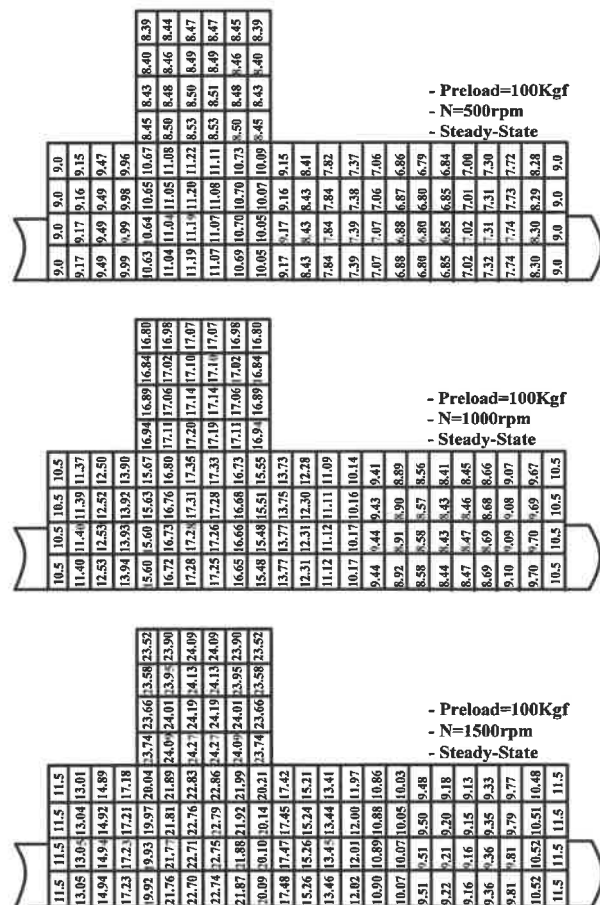


FIGURE 4. Differential Temperature Distribution of the Ballscrew System.

axial location in the radial direction are almost the same in the nut and shaft as shown in Fig. 4. This is due to big slenderness ratio of the ballscrew. To estimate preload change owing to the differential thermal growth between the nut and shaft elements on the groove, we simplify temperatures in the axial direction as a mean temperature of the radial ones according to nodes along the axial direction. For the j -th element, thermal expansions of the nut and shaft are given by

$$\begin{aligned}\delta_{th,j}^N(x) &= \sigma \Delta x T_j^N \\ \delta_{th,j}^S(x) &= \sigma \Delta x T_j^S\end{aligned}\quad (6)$$

where T_j^N and T_j^S are the nut and shaft mean temperatures of the j -th element, respectively. Then, the preload change due to the j -th differential deformation is given by

$$F_{p,j}^c = \frac{|\delta_{th,j}^N - \delta_{th,j}^S|}{\Delta x} A_{S,N} E_{S,N} \quad (7)$$

where $A_{S,N}$ and $E_{S,N}$ are the cross-sectional area sum and Young's modulus of the nut and shaft elements at the j -th location, respectively. Preload change of the ballscrew nut due to the thermal effect is obtained from the mean of $F_{p,j}^c$ along the contact surface of the nut and shaft.

Nut Stiffness Model with Thermal Effect

Adding the preload change obtained from the previous section to the previously developed nut stiffness model [1], the ballscrew nut stiffness

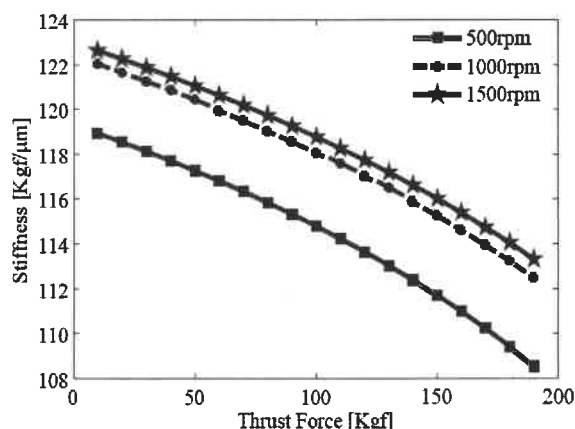


FIGURE 5. Nut Stiffness vs. Rotational Speed in the Double-Nut Ballscrew with Preload 100Kg.

including thermal effect is obtained according to load and assembly conditions. Fig. 5 shows thermal stiffness of the nut section according to rotational speed of the screw shaft. The increase in the rotational speed further increases preload differential of the ballscrew. It confirms that thermally induced preload increases the nut stiffness.

CONCLUSIONS

A nut stiffness model including thermal effect is proposed to develop an accurate stiffness model of a ballscrew system. Differential temperature distribution is estimated through the finite difference model and proper boundary conditions obtained from the drag torque and operating conditions. Preload change due to the differential growth between the nut and shaft has been obtained from the differential temperature. Applying the preload change to the previously developed nut stiffness model, thermal stiffness is obtained. It is confirmed that the stiffness increases as the differential growth increases according to the rotational speed.

ACKNOWLEDGEMENTS

This research is funded by Ministry of Knowledge Economy, Korea.

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C-LVDT Toolset Station for uses in Diamond Machining

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INTRODUCTION

The purpose of this project was to develop a device to measure the relative tool position of multiple diamond machining tools for the manufacturing of multiscale optics. Multiscale optics are biomimetically inspired. For example Moth's eyes, seen in Figure 1, with an overall aspheric shape on the order of 1 mm in diameter has small lenslets on the order of 10 micrometers in diameter and on those lenslets are subscale curved antireflective features less than one micrometer in size.

Such optics allow wide field of view and multifunctional capabilities not presently

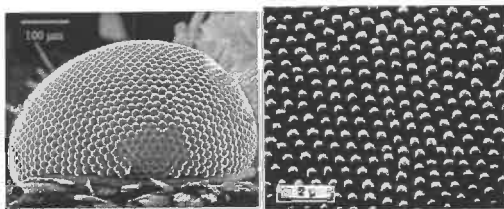


Figure 1: (Left) Compound moth eye ([1]). (Right) Scanning electron microscope image of fine detail on surface of moth's eye showing anti-reflective subwavelength surface structures ([2, 3])

manufacturable. Mechanical multi-axis diamond machining offers a potential method for manufacturing such structures.

To test this hypothesis requires that the position of the tip of diamond tools in the X, Y, and Z machine coordinates be determined with an uncertainty less than the wavelength of light. The targeted uncertainty for the project was to measure the relative tool position with a combined standard uncertainty less than ± 100 nm. Toolset stations using LVDT probes have been the traditional method of determining tool positions in diamond machining. Standard LVDT probes have thermal errors associated with heating of the inductive elements. In this project, a toolset station is developed with Lion Precision C-LVDT's that operate with an air bearing plunger against a precision capacitance probe and thus eliminate the inductive heating problem. A toolset station such as the one described in this work is a prerequisite for

manufacturing complex optics spanning many orders of magnitude. It would also have other uses such as tracking tool wear.

DESIGN

The Toolset Station device consists of two Lion Precision C-LVDTs, probe mounts, and an adaptor to connect the main spindle to the back of the probe mount. The adaptor has been manufactured with three pairs of rods acting as three grooves and the probe mount has three steel balls in order to create a three ball, three groove kinematic mount. A kinematic mount is used to reduce stresses in the system created by thermal expansion or over constraints. Also, the kinematic mount is used to reduce the contact area when mounting the toolset station to reduce settling time due to trapped fluid in the contact areas ([4]).

FUNCTION

The toolset station is mounted on the main spindle of a Moore Nanotech 350FG. The device uses two C-LVDT's: one probe mounted parallel to the Z-axis, the other probe capable of rotating in the X-Z plane. The Z-axis probe is used to measure the Z-axis position of the tool only. The rotating probe will be fixed at an angle, θ (see Figure 4), to the Z-axis which can be measured using a dial indicator. The angle θ will depend on the included cutting angle of the tool in order to have the greatest angle from the Z-axis. The greater θ can be the lower the errors will be in measuring the X-axis position of the tool. Also, the rotating probe can be set at 90° from the Z-axis and then the toolset station can be rotated using controlled C-motion in order to position the probe parallel to the Y-axis. By positioning a tool parallel to the Y-axis, the height of the tools can be determined.

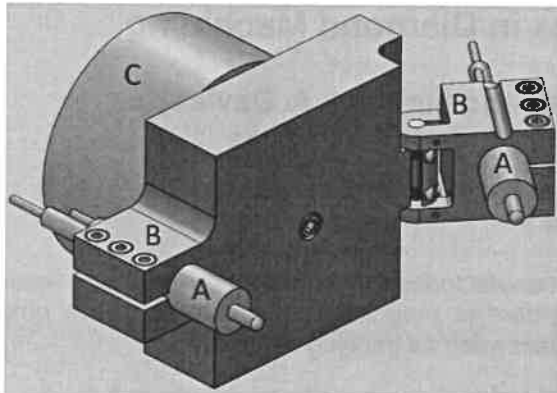


Figure 2: SolidWorks Solid model of the C-LVDT Toolset Station. (A) C-LVDT probes, (B) Probe Mounts, and (C) Spindle Adaptor

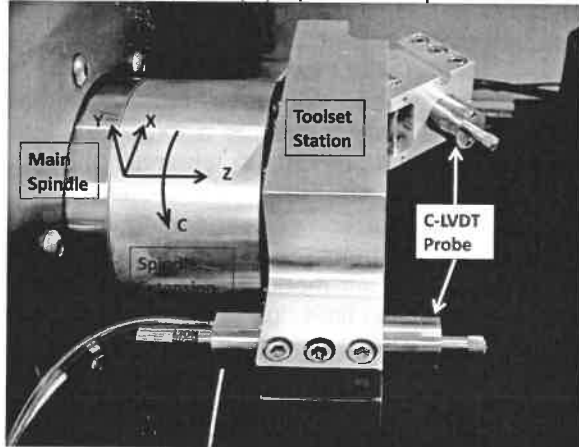


Figure 3: The toolset station mounted on the Moore 350 FG's main spindle.

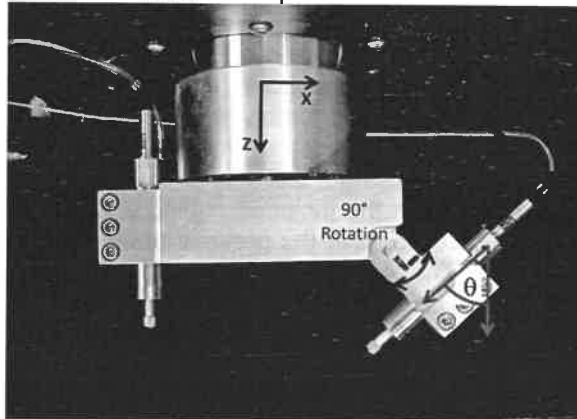


Figure 4: Top view of the toolset station showing the rotating probe's angle to the z-axis

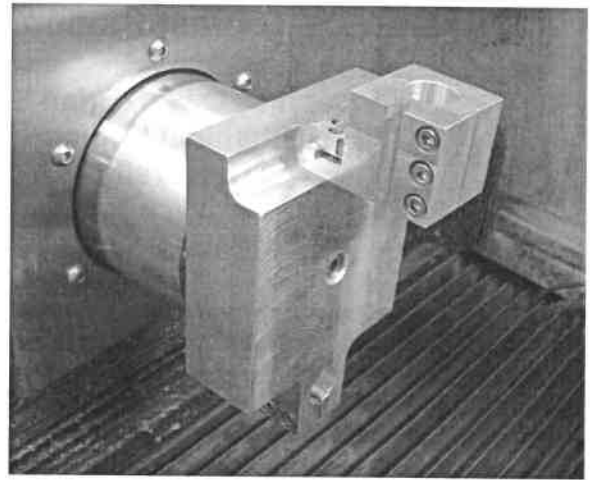


Figure 5: View of the toolset station with the rotating probe clamp set to measure the tool height

RESULTS

Preliminary repeatability testing was conducted to determine how well the tip of the tool can be positioned on the tip of the probe. A single probe was used, oriented in the Z-axis. Moving the tool across the probe in the X-Y plane until the probe measured a maximum was the initial method for finding the tool tip. This technique was useful for finding the tool tip quickly, however the resolution was coarse when near the tool tip. A second method was used to better find the tip. The second method began by moving the tool 50 microns to one side of the probe in the X-axis and the probe's measurements were recorded. The next step was to move the tool across and past the center of the probe. The tool was then moved until the probe measured the same value as the first side. The middle of the two X-axis positions became the recorded tool tip. The procedure was then repeated in the Y-axis.

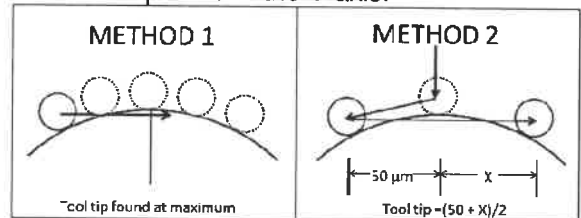


Figure 6: Diagram of the two methods used for finding the tool tip

Figure 7 shows that the tool tip was found with a standard deviation of about 5 microns. Using a 3 mm radius probe and a 1 mm radius tool, a tool positioned 5 microns off center results in an error of <10 nm which is near the sensitivity of the probe. The Z-axis position of the tool tip varied by ± 200 nm. The Z-axis position variation was dominated by thermal expansion due to the use of an aluminum prototype instead of a more stable steel or invar.

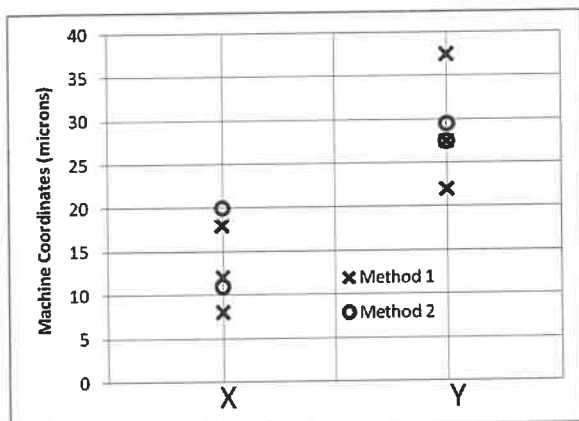


Figure 7: Recorded machine coordinates after 5 attempts to find the tool tip, using both procedures. The zero position on the plot was an arbitrary tool offset saved in on the machine to bring the recorded numbers closer to zero.

We will report on initial tests aimed at manufacturing tests components using this toolset station. Results will be reported on the following test conditions.

1. Thermal stability of the tool set station
2. Two nominally identical round nosed diamond tools will be located relative to one another. A flat will be started with one tool and continued with the second and the errors in the locations of the two surfaces will be reported.
3. One round nosed tool and one deadsharp tool will be located with the tool set station. A flat will be cut with the round nosed tool and lines will be ruled on the flat with the deadsharp tool and the results will be measured and reported.

ACKNOWLEDGEMENTS

We gratefully acknowledge the National Science Foundation (CMMI Grant #0927621) for their financial support.

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UNCERTAINTY MODELING OF BALLSCREW NUT STIFFNESS

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ABSTRACT

Uncertainty estimation of the nut stiffness of a ballscrew is conducted through Monte Carlo method. Expanded uncertainty is evaluated by stochastic analysis of the previously developed stiffness model with uncertainty effects. The results are verified through experiments.

INTRODUCTION

Ballscrews are widely applied to industrial machinery and precision machines. Positioning accuracy of a ballscrew system depends upon geometric error and axial stiffness of a feed drive system. Using NC (numerical control) function, repeatable geometric errors are compensated for easily. However, as the nut stiffness of a ballscrew system depends upon load conditions, geometric characteristics of the groove and assembly conditions, the process of estimating the nut stiffness is complex and should be conducted accurately.

The nut stiffness depends upon the Hertzian deformation between balls and grooves, and the body deformation of the screw shaft and nut. It is also varied according to the thrust force, groove characteristics and preload [1]. Fig. 1 shows contact state of the i th ball and groove subjected to thrust force F_t . In the figure, α represents contact angle between the ball and groove, and D is ball diameter. F_i denotes ball contact load, d_m means pitch diameter, r_s and r_n are curvature radii of the screw shaft and nut, respectively.

Using the given parameters, Hertzian constants between balls and grooves are derived through elliptic integrations [1-2]. Applying for the force equilibrium and compatibility conditions, the body deformation is estimated according to the assembly and load conditions of a ballscrew [1]. However, the conformity factor and the contact angle affect Hertzian deformation significantly. Since the conformity factor expresses the ratio between curvature radii of the screw groove and ball radius, groove error varies the contact angle

and the conformity factor. These values affect the variation of contact deformation. These uncertainties generate uncertainty of the contact deformation estimate. Uncertainty of effective areas of the nut and the shaft increase uncertainty as well.

In the uncertainty analysis, GUM (guide to the expression of uncertainty in measurement) with sensitivity coefficients has been widely used so far. However, uncertainty evaluation by GUM is improper for nonlinear models and systems with correlated uncertainty parameters [3]. Contrary to GUM, Monte Carlo method (MCM) is applied to nonlinear models with uncertainty. Correlation analysis is also possible through MCM. In this paper, uncertainty estimation of the nut stiffness model is performed according to uncertainty factors using MCM. Expanded uncertainty of the stiffness model is estimated by stochastic analysis of uncertainty effects. The modeling result is verified through experiments on the developed stiffness measurement system.

UNCERTAINTY MODELING

Many factors generate uncertainty of the previously developed ballscrew nut stiffness model [1]. There are ball diameter D , contact angle α , conformity factor $f_{s,n}$ and effective area $A_{s,n}^{eff}$, Young's modulus $E_{s,n}$ and Poisson ratio $\zeta_{s,n}$ of the shaft and nut, respectively. In the ball

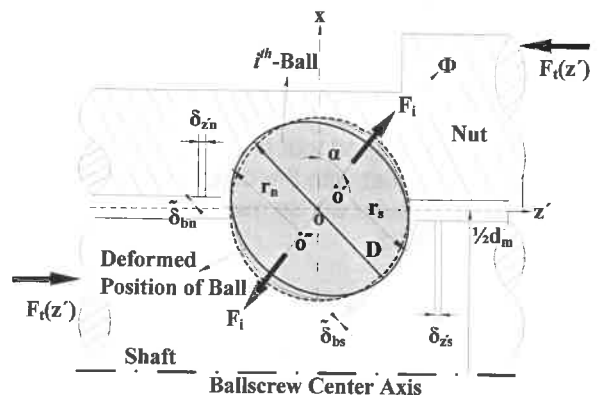


FIGURE 1. Ballscrew Geometry and Contact Mechanism.

fabrication process, ball diameter D has out-of-sphericity error. This changes the contact angle, conformity factors and effective areas. Material properties such as Young's modulus and Poisson ratio are also varied lot by lot in the manufacturing process. These effects generate uncertainty of the nut stiffness. Inaccuracy of the model generates the uncertainty as well. In this paper, parametric uncertainty due to the geometric and material properties is studied for the uncertainty modeling.

Fig. 2 shows GUM steps. In this case, input variables and corresponding sensitivity factors should be independent each other. For the correlated input cases, the sensitivity estimation is difficult in the GUM application. In such cases, MCM is a useful tool for uncertainty evaluation. Using MCM, we can handle many kinds of the input uncertainty as random numbers. As the sensitivity coefficients are not essential in the MCM, input covariance variables are computed automatically [5]. In this paper, we use MCM shown in Fig. 3 for the uncertainty estimation.

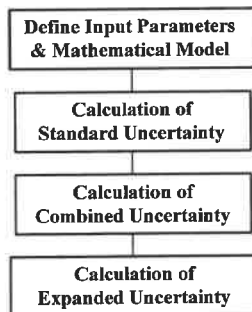


FIGURE 2. Uncertainty Evaluation by GUM.

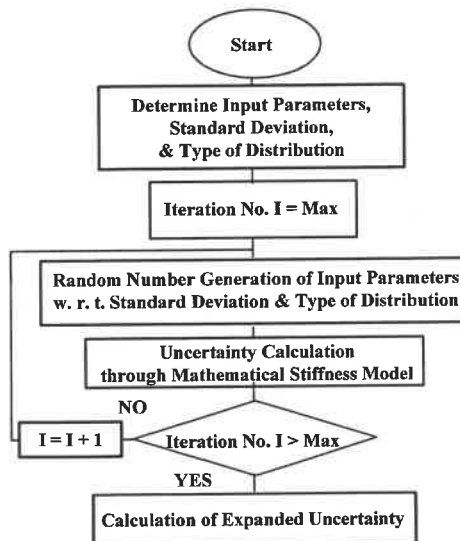


FIGURE 3. Procedure of Monte Carlo Method.

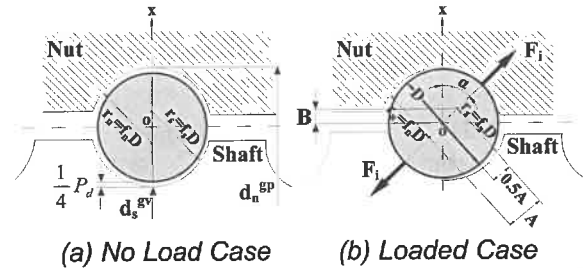


FIGURE 4. Geometric Relationships between Ball and Grooves.

Input Parameters

Table 1 and 2 show nominal input variables and parameters obtained from THK's double-nut ballscrew BNFN2505-5 for the stiffness model [1]. In this case, geometric input variables are correlated each other as described in previous section. To compute uncertainty by MCM, correlation between input variables should be calculated first through standard deviation analysis of input variables. As the ball diameter varies the contact angle and conformity factor, standard deviations of them are computed according to the standard deviation of the ball diameter. Out-of-sphericity error of a precision ball is $0.1\sim 1.6\mu\text{m}$ [4]. Radius of curvature of the ballscrew groove $r = 1.794\text{mm}$ for BNFN2505-5. Considering the contact geometry shown in Fig. 4, change of the contact angle is derived as given by Eqs. (1) and (2). In this case, A and B are temporary geometric parameters. f is a conformity factor. Diametral clearance C is assumed as 0.5mm [2]. Effective area $A_{s,n}^{eff}$, Young's modulus $E_{s,n}$, and Poisson ratio $\zeta_{s,n}$ are selected properly.

$$f = \frac{r}{D} \quad (1)$$

$$P_d = d_s^{gv} - d_n^{gp} - 2D$$

$$A = r_s + r_n - D = (f_s + f_n - 1)D = (2f - 1)D$$

$$B = \frac{1}{2}A - \frac{1}{4}P_d + \frac{1}{4}C$$

$$\cos \alpha = \frac{B}{0.5A} \quad \text{where, } C: \text{Diametral Clearance}$$

$$\alpha = \cos^{-1} \left(1 - \frac{P_d + C}{2A} \right) \quad (2)$$

Generation of Random Inputs

According to ballscrew fabrication characteristics in manufacturing processes, uncertainty factors, f , α , D , $A_{s,n}$, $E_{s,n}$ and $\zeta_{s,n}$ have normal distribution property. Using the Box-Muller method given by Eqs. (3), random variables with normal property are given by

$$Z_1 = \sqrt{-2\ln U_1} \cos(2\pi U_2)$$

$$Z_2 = \sqrt{-2\ln U_1} \sin(2\pi U_2) \quad (3)$$

U_1 and U_2 are uniformly distributed random variables between 0 and 1. Z_1 and Z_2 are normally distributed random values for the simulation. In this paper, 1000 random samples shown in Fig. 5 are applied for the Monte Carlo simulation of the nut stiffness.

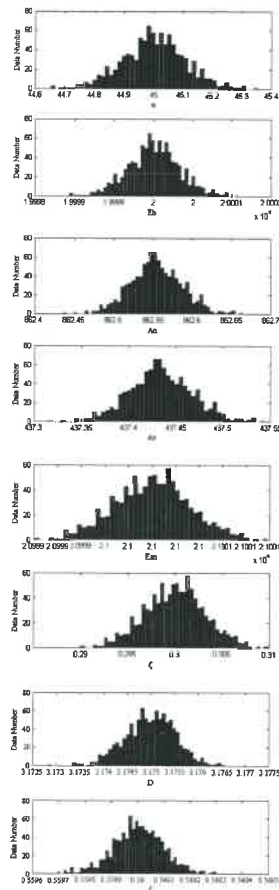


FIGURE 5. Normal Uncertainty Input Factors.

TABLE 1. Nominal Input Variables of BNFN 2505-5 for Uncertainty Analysis.

Ball Diameter D [mm]	3.175
Conformity Factor f	0.565
Contact Angle α [°]	45
Effective Area of Shaft A_s [mm ²]	437.435
Effective Area of Nut A_n [mm ²]	862.556
Young's Modulus of Shaft & Nut $E_{s,n}$ [Kgf/mm ²]	21,000
Young's Modulus of Ball E_b [Kgf/mm ²]	20,000
Poisson Ratio $\zeta_{s,n}$	0.3

TABLE 2. Geometric and Loading Parameters of BNFN2505-5.

Thrust Force F_t [Kgf]	150, 200, 250
Preload F_p [Kgf]	128.85
Shaft Groove Valley Diameter d_s^{gv} [mm]	22.2
Nut Groove Peak Diameter d_n^{gp} [mm]	29.3
Ball Center Diameter d_m [mm]	25.75
Total Ball Number Z	240
Ball Number per Pitch z	24

Stiffness Analysis by MCM

Table 1 shows nominal input variables of THK's double-nut BNFN2505-5 for the previously developed nut stiffness model [1]. For tensile preload 128.85Kgf, uncertainty estimation by MCM has been conducted according to the CC assembly [1] and load conditions given in Table 2. Table 3 and Fig. 6 show uncertainty evaluation results of the nut stiffness according to 1000 input uncertainty factors shown in Fig. 5. Mean stiffness of 82.78Kgf/ μ m is obtained for thrust force of 250Kgf. Estimated nut stiffness is obtained between 59.70 and 105.86Kgf/ μ m with 95.46% confidence.

TABLE 3. Uncertainty Estimation by MCM.

Thrust Force [Kgf]	150	200	250
Standard Uncertainty [Kgf/ μ m]	11.21	9.024	11.54
Expanded Uncertainty [Kgf/ μ m]	22.42	18.05	23.08
Theoretical Mean [Kgf/ μ m]	87.47	85.72	82.78
Confidence Level	95.46%		

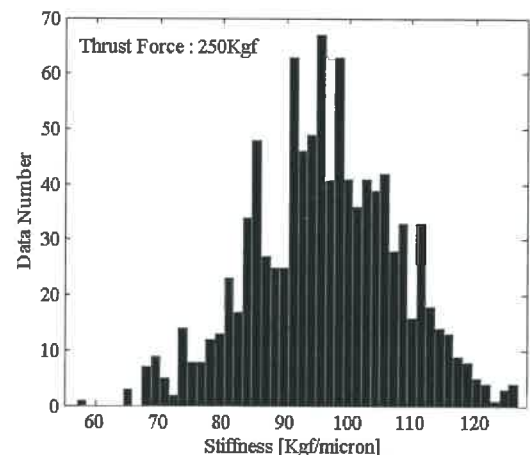


FIGURE 6. Stiffness Estimation through MCM (Thrust Force = 250Kgf).

EXPERIMENTAL VERIFICATION

To verify the uncertainty obtained through the MCM, 90 stiffness measurement experiments have been conducted on the stiffness measurement system shown in Fig. 7, which is equipped with a hydraulic load unit, two gap sensors and a LabView data acquisition system. Nut stiffness measurement experiments are performed for THK's BNFN2505-5 double-nut ballscrews with the CC assembly [1] and the load condition given in Table 2. According to Monte Carlo simulation, measured mean nut stiffnesses are obtained as 67.6, 65.2 and 63.4Kgf/ μ m for 150, 200 and 250Kgf thrust forces, respectively. Fig. 8 shows measured stiffnesses according to thrust forces. It is confirmed that experimental means are 76% of them obtained from the stochastic analysis of the MCM. Table 4 and 5 shows uncertainty measurement results. As the analytic uncertainty obtained through MCM includes experimental uncertainties, we can confirm the uncertainty evaluation by MCM.

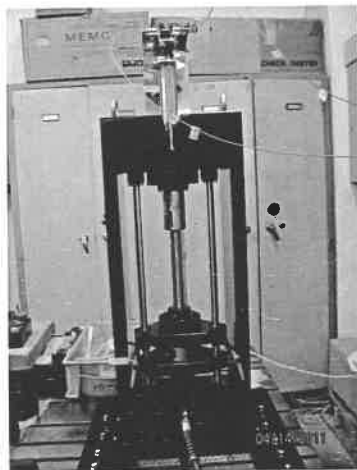


FIGURE 7. Nut Stiffness Measurement System.

TABLE 4. Uncertainty Measurement Results.

Thrust Force [Kgf]	150	200	250
Combined Uncertainty [Kgf/ μ m]	9.68	9.1	8.7
Expanded Uncertainty [Kgf/ μ m]	19.4	18.3	17.4
Confidence Level	95.46%		

TABLE 5. Confidence Interval of Measurements.

Thrust Force [Kgf]	Confidence Interval [Kgf/ μ m]
150	48.2 $\leq$ 67.6 $\leq$ 87.00
200	46.9 $\leq$ 65.2 $\leq$ 83.50
250	46.0 $\leq$ 63.4 $\leq$ 80.80

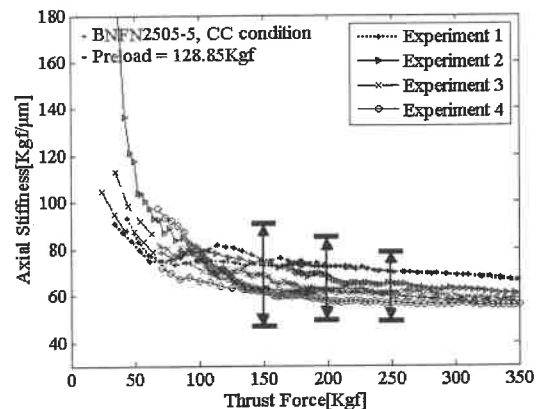


FIGURE 8. Stiffness Measurement Results.

CONCLUSIONS

Uncertainty evaluation of the nut stiffness model of ballscrews has been conducted through Monte Carlo method. To confirm the estimation, experimental verification is performed on the developed nut stiffness measurement system. It is evaluated that the experimental mean stiffnesses are 76% of the analytic mean estimates. To increase uncertainty estimation accuracy, modeling inaccuracy of the previously developed stiffness model should be studied in the future uncertainty analysis.

ACKNOWLEDGEMENTS

This research is funded by Ministry of Knowledge Economy, Korea.

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THE THREE-BEAD KINEMATIC COUPLING

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Abstract

As part of a concentrating photovoltaic solar power project, there was a requirement to create a ton-capable ruggedized kinematic coupling to accommodate temperature differentials and to enable fast field replacement of power heads, at a solar-acceptable price point. This mount is described in this paper.

The kinematic coupling comprises three conical depressions and three beads, the beads being partially spherical bodies with specially formed holes, capable of travelling on shafts that lie in the plane of the coupling and whose axes intersect approximately at its center.

The components of the coupling are comprised almost exclusively of turned features and have no flat-ground surfaces. The coupling is preloaded to allow operation at any attitude, and keyed to prevent accidental rotation.

The Three Groove Mount

A classic three groove mount is a commonly used single-faced kinematic mount consisting of three v-grooves on component A, and three fixed spheres on component B.

If the grooves and spheres are considered ideal, the spatial relationship between components A and B is kinematically defined when each sphere is tangent to both walls of its v-groove.

The load capacity of a three groove mount is limited by Herzian yielding at the points of contact between the spheres and the v-grooves. This limit is often too strict for field applications, and even if using large connectors, the mating surfaces are easily plastically deformed.

First attempt: The Spherolinder Mount

The Spherolinder mount was developed in 2004 by g2 Engineering as a way to increase the load capacity of the three-groove mount.

By replacing the bottom half of the interface ball with a cylinder that is co-axial with the spherical half, the mediating body is able to kinematically interface with both a V-groove and a conical depression, mimicking the behavior of the tooling ball, but having only line contacts.

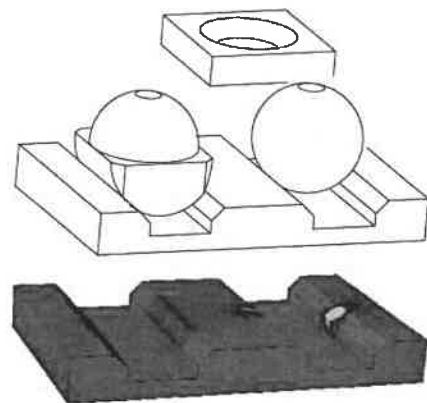


FIGURE 1. Single load point of a Spherolinder kinematic mount and load distribution in comparison to a sphere.

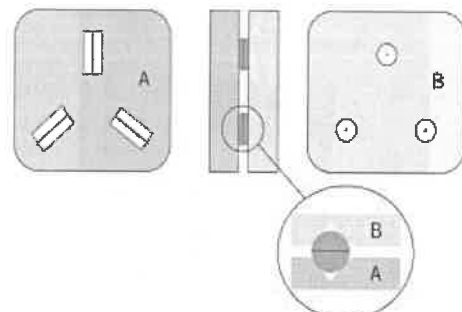


FIGURE 2. Spherolinder kinematic mount – Three grooves roughly aligned towards a common center.

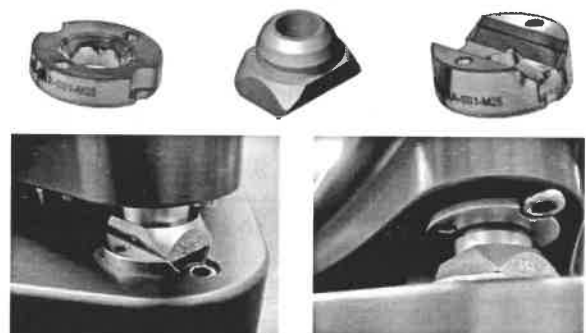


FIGURE 3. Product implementation of Spherolinder kinematic mount.

As before, it is assumed that the geometrical quality of the mediating surfaces is much higher than the tolerances of the overall part.

The Herzian load capacity of the line contact in this configuration is about 100x that of a three-groove mount, and the radius of curvature at the point of contact remains the same, preventing the entrapment of particulates there.

Fabrication of the Spherolinder requires that both the cylindrical surface and spherical surface be fabricated in the same machining setup, which required the design of a custom grinder.

The Spherolinder is an existing product sold by g2 Engineering. It performs well, but is difficult to fabricate and is too expensive to be used in a solar application.

Second attempt: The Three Bead Mount

The three bead mount is a second attempt at a ruggedized kinematic mount that reduces cost relative to the Spherolinder but is comparable in performance.

In each of the three bead matings, we replace the v-groove and cylindrical surface features of the Spherolinder with a hard centerless ground pin and modified hole pattern ("the snowman hole".)

The sphere-on-a-shaft is termed a "bead".

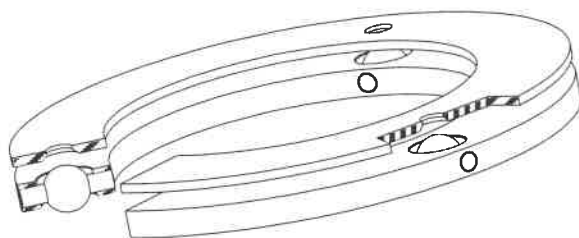


FIGURE 4. An idealized Three Bead Mount.

Since a round shaft inside a round hole is not a well defined mating, we introduce a modified hole pattern to replace the round hole and address this shortcoming.

The snowman hole (denoted SM in Figure 5 below) creates a functional V-groove without having to grind flat surfaces, reducing the fabrication cost

Since the new V interface is inverted and resides inside the ball, the stack height is greatly reduced as well.

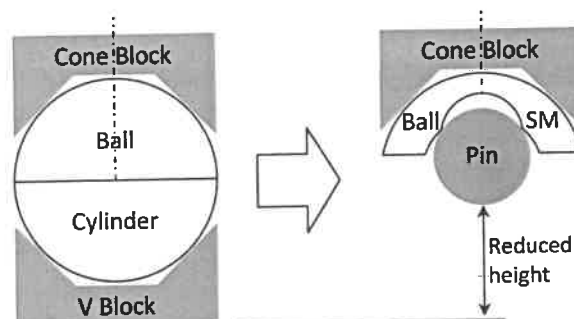


FIGURE 5. A comparison of the Spherolinder mate to the bead mate.

The new concentricity requirement is between the bored hole and the spherical surface, which is relatively easy to achieve by fabricating the bead on a live-tool lathe, again in a single setup.

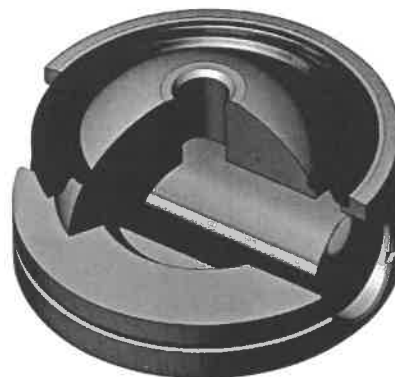


FIGURE 6. Product implementation of the Three Bead Mount (cone block not shown).

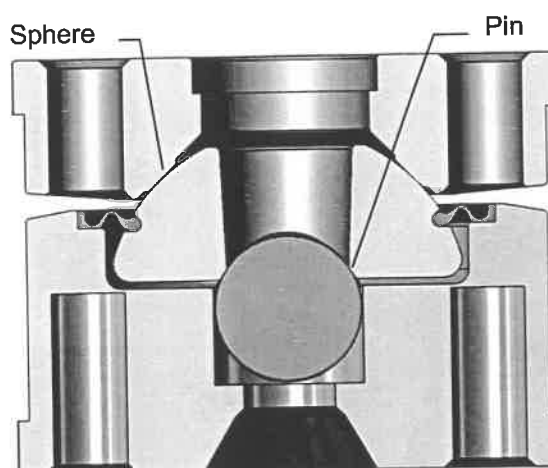


FIGURE 7. Cross section showing two contact surfaces (sphere/cone, and pin/snowman hole)

Load capacity

The bead interface and the Spherolinder are very similar in terms of calculating load capacity, except that the Spherolinder's optimal geometry had the cylinder length equal 1.1 times the diameter of the sphere, whereas in a bead interface it is in fact somewhat shorter than the diameter, resulting in about half the load capacity.

This is more than offset by the ease of fabrication, which allows a small increase in diameter to compensate for this decrease in performance.

Status

A first set of prototypes is under fabrication, and test results are expected by November 2011.

PRECISION WEAR INSPECTION OF SINGLE-CRYSTAL DIAMOND TOOL USED IN ELECTROLESS NICKEL TURNING

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INTRODUCTION

Single-point diamond turning (SPDT) is excellent optical fabrication technique [1] and is easy to generate aspheric shape compared to a conventional grinding and polishing processes. Single-crystal diamond tools are widely used in the SPDT. The machined surface in terms of shape and surface roughness critically depends on the machine tool and the single-crystal diamond tool used in the fabrication process.

Generally, non-ferrous materials and plastics can be used in the SPDT [1]. The well-known metal materials are aluminum alloys, oxygen free copper, and electroless nickel that is an alloy of nickel and phosphorus. In material point of view, the surface topography after the SPDT depends on the some material aspects such as purity, grain size, and anisotropy [2, 3]. Thus, the material is very important to obtain super-smooth surface. In this sense, the electroless nickel of the amorphous structure is one of excellent materials in the SPDT. In addition, electroless nickel has been widely used in industrial and optical applications [4]

Diamond is one of the hardest materials. However, the wear of the diamond is avoidable during SPDT [5, 6]. The tool wear decreases surface quality. Thus, it is important to understand wear of the single-crystal diamond tool to improve machined surface quality.

In this paper, we deal with inspecting the wear of a single-crystal diamond tool using laser scanning microscope after SPDT.

EXPERIMENTAL SETUP

A diamond turning machine tool used in this experiment has two perpendicular slide tables (Z and X axes), an air bearing spindle on the Z-slide table, and a rotary table (B-axis) with a tool

poster. A laser interferometer system used to determine accurately positions. X and Z tables are controlled within 1 nm positional step. The machine tool was set up within a vibration isolator and supported by air mounts to prevent external vibration. The environment is maintained by a clean room of the class 100 and controlled within temperature of 0.1°C.

Electroless nickel was plated on the aluminum alloy, A5052, of 45mm diameter. The thickness of the electroless nickel contained about 10% weight phosphorus was 110 µm.

A single-crystal diamond tool of a nose radius of 5 mm that the crystallographic plane (110) was rake face was used, and rake and clearance angles were 0 and 5 degrees, respectively, and the window angle was 60 degrees. The shank of the diamond tool was made of steel.

The single-crystal diamond tool after SPDT was inspected with a confocal laser scanning microscope (LSM). The wavelength of a semiconductor laser used in the LSM was 408±5 nm, and the height scan resolution was 50 nm. Magnifications of the LSM were 5X, 20X, and 100X. The system provided several analysis tools, for example measurements of angle, length, and so on. The LSM was coupled with a scanning probe microscope (SPM), atomic force microscope. The detail observation was also possible by the SPM after inspection using LSM.

MACHINING OF ELECTROLESS NICKEL

Before SPDT of the electroless nickel using the single-crystal diamond tool, a poly-crystal diamond tool was used to make the workpiece be flat. It promised that the electroless nickel was fully cut when the first machining was carried out using the single-crystal diamond tool.

In addition, it make easy to calculate the machined distance during the SDPT.

Machining conditions of the machine tool were 1000 rpm in spindle rotation, 2 $\mu\text{m}/\text{rev}$ in feed rate, and 1 μm in depth of cut, and were not changed during machining of the electroless nickel workpiece using a single-crystal diamond tool of 5 mm in radius. The machined distance per one pass was 795.2 m. The machining of electroless nickel was done during four passes, and the total distance was 3.18 km. The machined electroless nickel was examined with the LSM and the SPM. Fig. 1 shows the surface topography measured with the LSM and the SPM. The root-mean-square surface roughness and the peak-to-valley (PV) were 12.9 nm and 53.97 nm, respectively. Regular pattern was observed and the shape of each groove was very similar.

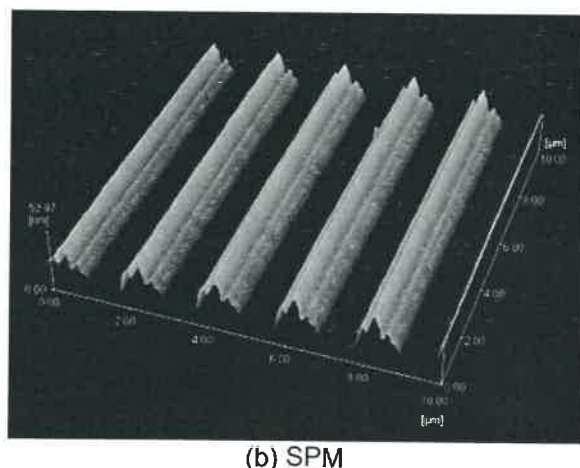
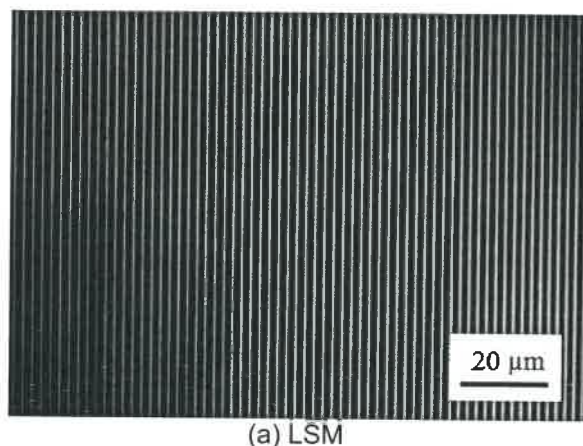


FIGURE 1. Machined surface of the electroless nickel. The groove pitch was 2 μm which corresponded to the feed rate in the machining condition.

TOOL WEAR INSPECTION

Before using the single-crystal diamond tool of 5 mm in radius, the nose radius was measured with the LSM. It was 4.948mm. The diamond tool was inspected again with the LSM after machined 3.18 km distance, after four passes. Before measuring the rake and frank faces, chip of the electroless nickel was clearly removed on the single-crystal diamond tool. Fig. 2 (a) shows the image of the worn single-crystal diamond tool of 5 mm nose radius. A defect, pointed out by a white arrow shown in the Fig. 2 (a), on the diamond tool was found. The defected was made an instant when the diamond tool attached on the electroless nickel for face cutting. When the diamond tool attached on the electroless nickel, it underwent very small oscillation which was very rapidly damped and kept a constant during cutting as shown in Fig. 3. It expected that the defect on the diamond tool did not give defect on the machined surface of the electroless nickel.

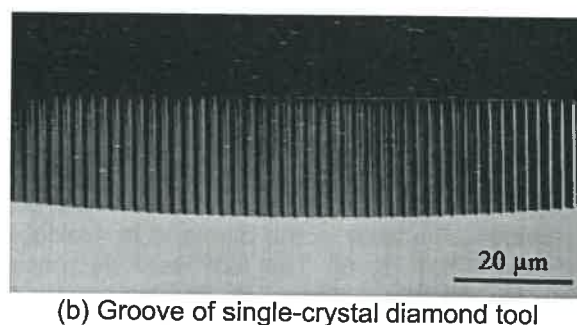
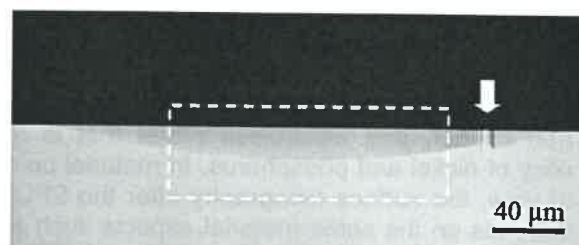
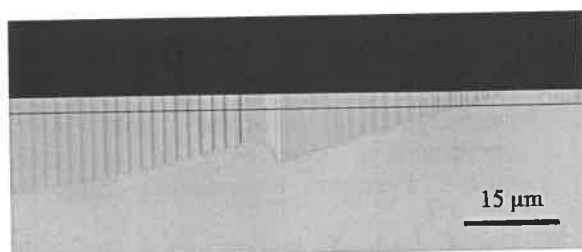


FIGURE 2. Frank face of the worn single-crystal diamond tool after machining electroless nickel of 3.18 km distance.

Fine grooves of 2 μm in width and 19.5 μm in maximum length on the worn diamond tool were clearly observed as shown in Fig. 2 (b) which corresponded to the rectangular part of the Fig. 2 (a). The interval of grooves was the same as the feed rate in the machining conditions. The groove depth was 50 nm. However, the groove

depth was difficult to confirm as 50 nm exactly because the depth resolution of the LSM was 50 nm. Thus, the frank face of the worn diamond tool was measured with SPM as shown in Fig. 4. The PV was 42.94 nm. If the groove on the diamond tool could exactly be copied on the machined electroless nickel surface, without considering groove deformation on the electroless nickel surface, the groove depth could be 42.94 nm. However, the machined surface of the electroless nickel showed slightly high PV value, 53.97 nm. It could explain that the cut point by the single-crystal diamond tool on the electroless nickel was sprung back after cutting, and the amount was approximately 10 nm.



(a) Defect on the diamond tool



(b) Line profile

FIGURE 3. Measured line profile for the defect on the diamond tool after electroless nickel turning. The line profile (b) corresponded to the line in (a).

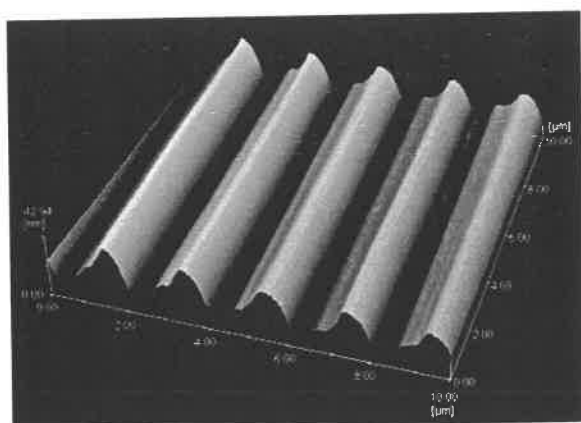


FIGURE 4. Frank face (measured with SPM) of the worn diamond tool after electroless nickel turning.

The clearance angle of the diamond tool could be measured with the LSM by measuring vertical profile of the frank face as shown in Fig. 5. The average value was 4.96 degrees (in five measurements) which was very close to 5 degrees which was designed value in making the diamond tool.

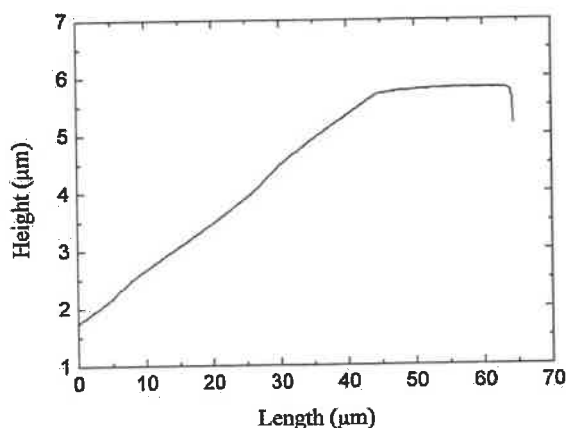


FIGURE 5. Measured profile of the frank face.

There was no damage signal on the rake face during machining of the electroless nickel in 3.18 km cutting distance. The cutting edge of the diamond tool was worn only. The measurement of the wear length on the rake face was not easy because the quantity was very small as shown in Fig. 6. Using the measured clearance angle and maximum groove length as shown in Fig. 5, the wear length on rake face could be calculated and it was 1.7 μm. The worn width as shown in Fig. 1 (a) was 285 μm. Using this length and the measured nose radius of 4.948 mm, the worn length could be expected and it was 2.01 μm which was very close to the measured value of 1.7 μm.

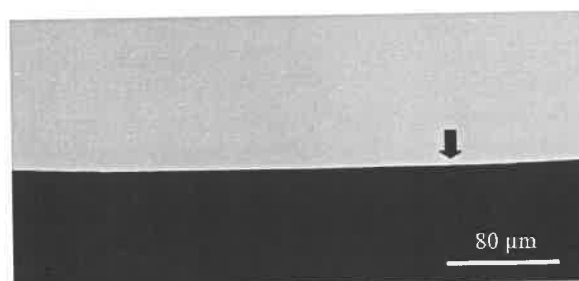


FIGURE 6. Rake face of the worn single-crystal diamond tool after machining electroless nickel in 3.18 km cutting distance. The worn part of the diamond tool looked like almost straight line.

The defect pointed out by a black arrow shown in Fig. 6 was clearly found. The step difference in the defect was $4.1\text{ }\mu\text{m}$ which was measured by LSM profile as shown in Fig. 3. The step difference was larger than the worn length of $1.7\text{ }\mu\text{m}$. It meant that the impact is relatively large when the diamond tool attached on the workpiece for cutting.

CONCLUSIONS

The wear of a single-crystal diamond tool after machining electroless nickel of 3.18 km in distance was inspected with the LSP and SPM. The fine grooves on the flank face and on the machined electroless nickel surface were observed clearly. Intervals for these grooves were $2\text{ }\mu\text{m}$ which was the same value of the feed rate. When the diamond tool attached on the workpiece, the impact gave rise to tool defect, and the step difference of the defect was $4.1\text{ }\mu\text{m}$. The worn length on the rake face could be estimated by measuring the flank angle and the groove length, and it was $1.7\text{ }\mu\text{m}$. The groove depth of the diamond tool was measured as 42.94 nm PV .

Wear of the single-crystal diamond tool are reflected on the machined surface. Thus, the wear information can be provided to improve surface quality, for example surface roughness and shape for fabricating x-ray microscope mirror. Using the LSM and SPM together, the wear information can be found. For example, the information of the wear length on the rake face, $1.7\text{ }\mu\text{m}$ in this study, can give hints to improve aspheric shape accuracy for long cutting distance.

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Face grinding monitoring and optimization through force measurements

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INTRODUCTION

High hardness and toughness are desirable mechanical properties of tungsten carbides (WC) which make them useful as wear resistant materials for cutting tools and die sets, etc. However, these same properties which make WC so useful for many applications in the manufacturing industry also make them difficult to machine into desired shapes.

Electric discharge machining (EDM) is one way to obtain a desired shape but the typical surface roughness is between 250 to 125 microinches. This makes EDM a good choice for acquiring a rough shape in the tungsten carbide material but a second process is needed for final precision surfaces.

Grinding with a diamond wheel can obtain much better surface finishes below to 8 microinches. However, the ability of the grinding process to produce the desired surface roughness for precision applications depends on many factors such as the rotational speeds of the work and wheel, the rate of feeds, the type and amount of coolant, along with the mechanical properties of the tungsten carbide.

Many parameters need to be controlled and affect the grinding process's ability to remove material and produce surfaces. Monitoring the grinding forces is a technique that can be used to diagnose a grinding process for quality control purposes as well as optimization of parameters.

For this work, the relationships between low amplitude force signals of precision face grinding to material removal rate, grinding wheel conditions, and coolant flow will be established. The displacement sensing spindle, shown in Figure 1, is calibrated to measure grinding force. Also a theoretical material removal rate is used to formulate a relative grinding intensity factor to optimize the process. The final step is to apply this knowledge to the micro end grinding process in order to push the very small tool to its limits before breakage. The material removal will be optimized with the greatest value of relative grinding intensity.

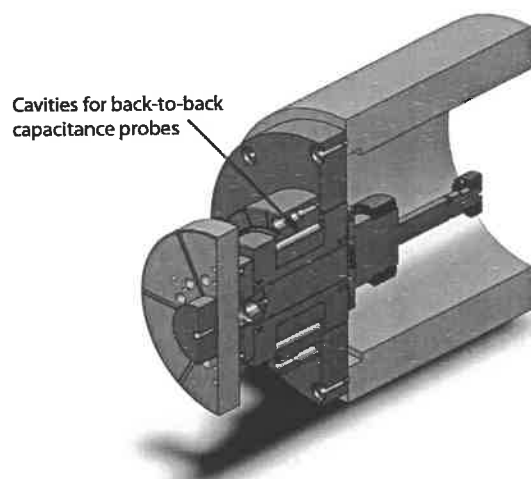


FIGURE 1. Instrumented work spindle with embedded capacitance probes measuring the time varying displacement between the rotor and stator during the grinding process.

Various types of process signals, data feature extraction methods, and detection algorithms have been developed for monitoring different material removal applications. Common process signals used for monitoring grinding are force, spindle load, acoustic emission, sound, structure vibration and hydrodynamic pressure. Among these, grinding force is usually considered as the most relevant for indirect and continuous monitoring of tool wear [1]. However, the grinding force can be used for more than just tool wear monitoring. Many factors influence the grinding force such as material removal rate and wheel conditions [2,3].

For this work, previous process models, novel signal conditioning suitable for detection of low amplitude force, and algorithms to identify various grinding phenomena in the force data will be used. Combining detection methods with an understanding of the interplay of factors in the process will then be applied to micro grinding so that material removal will be optimized with the greatest value of relative grinding intensity.

MACHINE SETUP

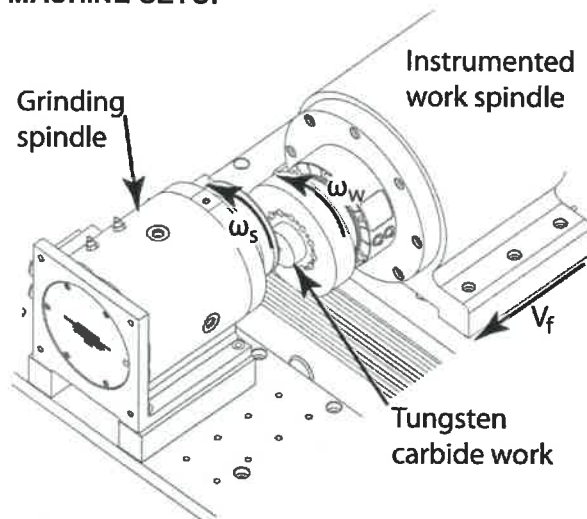


FIGURE 2. CNC grinding machine setup with displacement-sensing spindle with tungsten carbide work calibrated to output grinding force.

In this work a rotation grinding configuration also known as Blanchard grinding, which is surface grinding using the face of the wheel, is used. Face grinding does differ from a majority of the published papers where the grinding was performed predominantly with the peripheral surface of the grinding wheel [4].

For this setup shown in Figure 2, the orientation and geometries, materials and rotational speeds of the work and wheel will all remain constant throughout testing. Only two variables can change which are; the axial infeed plunge rate and depth of cut. Using these inputs the material removal rate and the process itself will be studied under the constraint of the low amplitude force measured from the work spindle.

THEORETICAL ANALYSIS

The diamond abrasive wheel cuts from the edge to the center of the tungsten carbide work. The work is fed horizontally in the axial direction into the wheel. The depth of cut is equal to:

$$a = V_f / \eta_w$$

Where a is the depth of cut (μm), V_f is the horizontal plunge infeed velocity ($\mu\text{m}/\text{min}$) and η_w is the work's rotational speed (rpm). The material removed from the workpiece is a cylinder and its rate is define as

$$Q_w = \pi r_w^2 V_f$$

Where Q_w is the material removal rate (MRR) and r_w is the radius of the tungsten carbide puck. As seen in the following equation for the MRR, Q_w is a function of the size of the work and the plunge infeed rate. Assuming, both wheel and work are rotating the material removal rate Q_w can be maximized by increasing the feed V_f of the work into the wheel. However, this is only a theoretical formula for the MMR and does not express some factors such as coolant, grit size, grit density, material types, etc. that will influence the MRR.

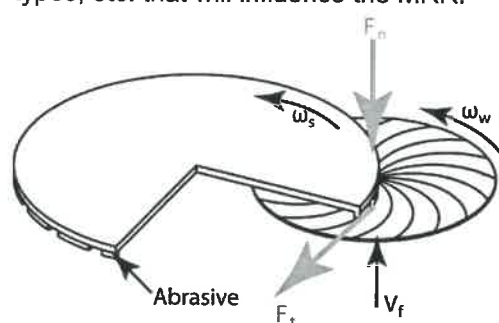


FIGURE 3. Rotary grinding process of tungsten carbide work with grinding forces shown.

Many of these factors can be combined into the specific grinding energy e_c (J/mm). The specific grinding energy is the energy that must be expended to remove a unit volume of work material. The specific grinding energy is directly proportional to the forces generated in the grinding process. For the geometry shown in Figure 3 the tangential force will be taken to be

$$F_t = \frac{e_c Q_w}{v_s}$$

Where v_s is the speed of the wheel (mm/min). The tangential and normal grinding forces are related by the grinding coefficient such that the normal grinding force is equal to

$$F_n = \frac{e_c Q_w}{\mu v_s}$$

The relative grinding intensity factor (RGI) is taken by solving for the inverse of the specific energy:

$$RGI = \frac{1}{e_c} = \frac{Q_w}{F_n \mu v_s}$$

Using the previous equation as a process indicator (based on the inverse of the specific energy) makes sense since the wheel work interaction must follow the law of energy conservation. The indicator can be utilized to measure the degree of wheel work interactions during grinding. A high RGI indicates a high efficiency with high material removal rate. Also the RGI can be used for monitoring the process, since a decreasing RGI with time and constant input parameter, probably means the grinding wheel needs redressing. This RGI will be calculated and displayed real time with the unique software developed in this work.

PRELIMINARY TESTING

The data from the 1V/ μ m capacitance probes (model #C2-C), targeting the back and front thrust plates measuring the time varying gap between the stator and rotor of the spindle, is collected from a National Instruments PCI-6032E multifunction analog, digital and time I/O card. The DAQ card is triggered from the 1024 count signal of the work spindle's encoder so that each revolution has the same number of samples.

Figure 4 shows an initial cylindrical face plunge grinding operation where the 1024 samples per revolution have been converted into degrees and the multiple revolutions stack together in time to form the following mesh plot. For this operation the measured displacements from the capacitance probes are a combination of both axial and tilt, and are calibrated to determine the grinding forces. A calibration factor of about 220N/ μ m was found using an Interface SM load cell.

As seen in the Figure 4, low amplitude force can be measured. The initial touch off was detected

by the spindle which corresponded to a grinding force below 2 Newtons.

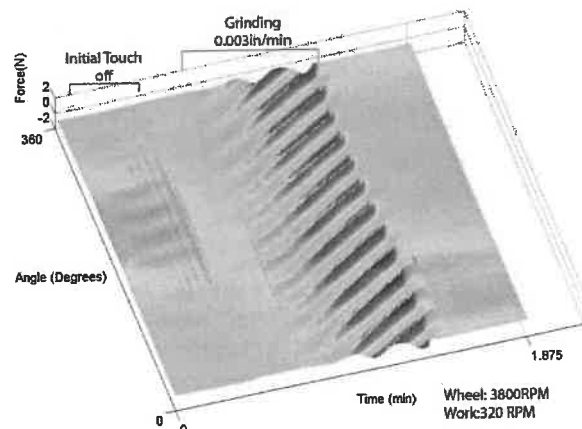
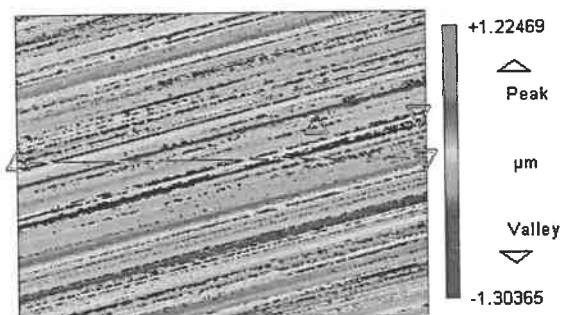


FIGURE 4. Mesh plot of cylindrical plunge grinding forces acquired from the instrumented work spindle

The surface generated from this initial grinding operation was measured on a Zygo NewView 5000 with a 10X objective, one of the measurements is displayed in Figure 5.

■ zygo



PV	2528.343	nm
rms	245.119	nm
Ra	194.633	nm
Size X	0.36	mm
Size Y	0.27	mm

FIGURE 5. Surface of WC work measured with Zygo NewView 5000.

As a comparison the WC ground workpiece was compared to a signal point diamond turned (SPDT) surface. The surface generated by the diamond wheel was achieved much more rapidly and had a higher quality finish. However, the wheel's surface did not break down as fast as the diamond tool.

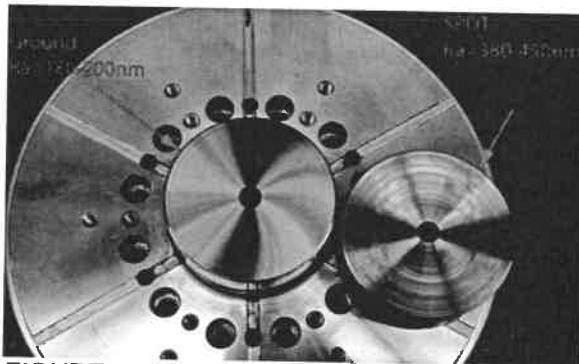


FIGURE 6. Comparing two WC pucks. The puck on the right was machined with a diamond abrasive wheel with a surface roughness between 180-200nm. The one on the right was machined using SPDT and has a surface roughness of 380-450nm with visible rainbow on the surface.

FUTURE WORK

It was found that the WC workpiece can be machined effectively with high quality surface finishes and the low amplitude forces at 2 Newtons detected.

Future progress for this work will continue in developing an appropriate signal conditioning suitable for processing low amplitude force signals below 2 Newtons from a very light end grinding process on WC workpieces, followed by creating algorithms to identify various grinding events in this force data.

The final goal is to apply this newly acquired knowledge to the micro end grinding process in order to push the very small tool to its limits before breakage, so that the material removal is optimized with the greatest value of relative grinding intensity achieved.

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MODELING THE POLISHING PROCESS WITH RESILIENT DIAMOND TOOLS FOR MANUFACTURING OF COMPLEX SHAPED CERAMIC IMPLANTS

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INTRODUCTION

Artificial joint implants create high demands on the manufacturing technology concerning surface quality and shape accuracy. With the application of brittle, hard, and wear resistant bioceramics (Al_2O_3 and ZrO_2) an increase in life span of hip implants up to more than 20 years could be achieved [1]. To get an economical process for more complex implants made from ceramic materials, e.g. a knee implant, new manufacturing strategies are required. Due to its material properties, grinding with diamond tools is the only way to machine oxide bioceramics with appropriate material removal rates [2]. To achieve high surface qualities, different precision machining processes were developed. Nonetheless, neither of these processes is appropriate for machining free form geometries such as knee implants; due to high influences on form accuracy [3].

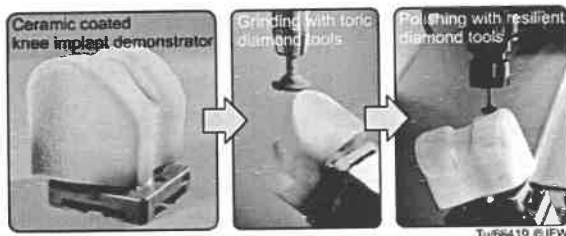


FIGURE 1. Developed processes for fully ceramic knee implants.

Therefore, a simplified process chain was developed which allows processing ceramic knee implants via one clamping on one machine tool (Figure 1). Firstly, the near net shape sintered and hipped ceramic body will be ground by toric diamond tools on a multi-axis machine tool to achieve high geometrical form accuracy [3]. In a second step, a resilient bond diamond tool is used to smoothen the roughness peaks of the ground surface. The tool path and force controlled polishing method keeps the influence on the shape accuracy at a minimum along with

lowest surface roughnesses ($S_a < 20 \text{ nm}$) (Figure 2).

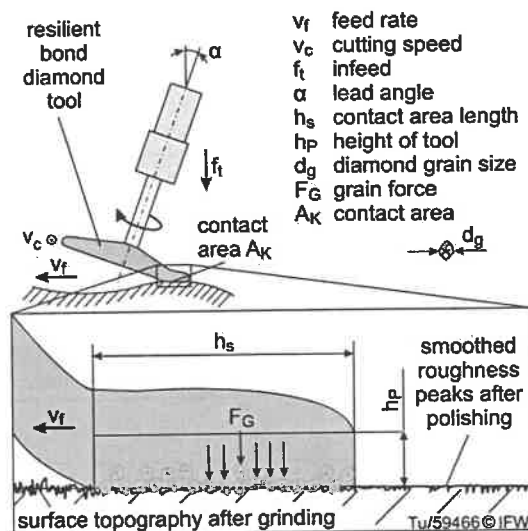


FIGURE 2. Geometric and kinematic parameters for the polishing process with resilient tools.

This paper follows up to the presented method from ASPE 2009 where the material removal mechanisms during polishing of ceramic as well as a basic modeling approach to determine the number of active grains were presented [4]. This approach is now expanded by calculation of the number of grains and verified by several experiments.

MODELLING THE POLISHING PROCESS

During polishing the removal of the roughness peaks depends on the number of grains N_{KP} sliding over a surface unit during a specific time unit and on the single grain force F_G (Figure 2). The number of active grains is governed by system parameters tool geometry and specification as well as process parameters feed rate v_f , cutting speed v_c , pressure/infeed f_t , tool path distance f_p and inclination α of the tool. All characters can be ascribed to the number of active grains and the grain force (Table 1, step 1). Assuming a regular grain distribution (2)

in the tool bonding and a contact area A_K between the tool and the surface the number of grains in the bonding volume over the supported area $N''_{KP,sec}$ can be calculated. Considering various grain distributions and grain shapes the number of active grains in the contact area N''_{KP} can be calculated (3). The model is specified by means of the process parameters cutting speed, tool path distance and the area which should be polished (4/5).

TABLE 1. Modeling steps.

	Parameter	Symbol	Description	Step
System parameter	concentration, grain volume, grain density	$N'''_{KP} = c_0$	grain density in bonding	(1)
	bonding vol., contact area, height of tool	$N''_{KP,sec}$	no. of grains in volume over supported area	(2)
	distribution, grain shape	$N''_{KP,A} / N''_{KP}$	no. of grains in contact area/ referred to area unit	(3)
Process parameter	cutting speed, feed rate	$N''_{KP,V}$	no. of grains per tool rotation	(4)
	path distance, polish area	$N''_{process}$	no. of grains polishing a spec. area in spec. time	(5)
	stiffness, number of grains (3)	F_G	single grain force	(6)
Work-piece	indentation depth and single grain force (6)	h_{cu}	chip thickness	(7)

The force acting in the contact zone can be determined by stiffness investigations. Using the number of active grains N''_{KP} and assuming that the normal force predominantly acts on the active grain contact, the single grain force F_G can then be calculated (6). The chip thickness can be determined by the material specific indentation depth (7).

MODEL VERIFICATION AND RESULTS

In order to develop a modeling approach to predict the material removal during the polishing process, the modeling steps 1-7 have to be verified. Conclusions about the grain distribution and shape (3) are evaluated by μ -CT (x-ray computer-tomography), SEM (scanning electron microscopy) and video-microscopy of the bonding structure (Figure 3). Figure 3a shows a voxel-based morphometry of the tool body. In contrast Figure 3b illustrates an x-ray photograph of the tool and its particles. The voxel-picture suggests a reinforcement of the diamond concentration C in the outside volumes of the bonding, especially for C200. A

quantification of the number of grains by the voxel-method is impossible due to low resolution of the μ CT (15 μ m - 20 μ m). However, the x-ray picture allows a nearly constant distribution of the grains. The percentage of white picture points increases with the number of grains (Figure 3b, bottom). The SEM- and video-photographs in Figure 3c and 3d were used to identify the grain shape of the diamond (type RVM-N, Diamond Innovations) and assign them to a calculable geometrical object. The typical, blocky grain can be matched to a double truncated pyramid.

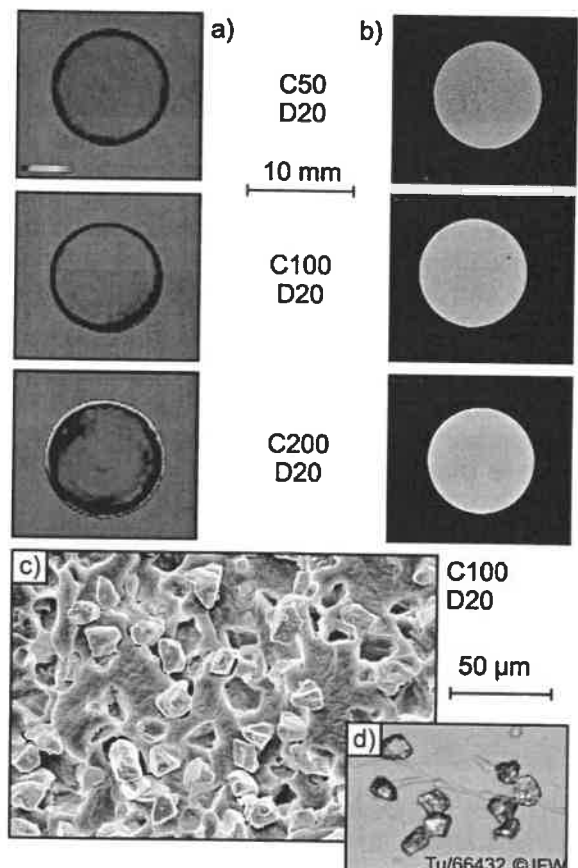


FIGURE 3. μ CT (a), x-ray (b), SEM (c) and video-microscope (d) of bonding structure and grain shape.

Assuming a regular distribution of the diamond grains, a predictable distribution structure has to be determined to describe the location of the grains in the bonding. This is required to calculate the number of grains $N''_{KP,A}$ (3). For this purpose, the crystal structures "cubic primitive lattice pcc", "body-centered cubic lattice bcc", "face-centered cubic lattice fcc", "cubic-diamond fcc" and the model by Yegenoglu [5] were compared. For each type of tool 108

grain shape: double truncated pyramid

distribution: cubic-diamond, (fcc)

manufacturer 1: manufacturer 2: bonding: silicone grain: RVM-N

10 cm

Tu8623 © IFW

Furthermore, the knowledge of the resilience of the tools is required to estimate the single grain force during the polishing process. To evaluate the dynamic stiffness, a rotational cutting force dynamometer is applied to measure the normal force in process. The stiffness c is calculated by the quotient of force and change in length (f_t). The reciprocal $1/c$ is the resilience. In Figure 5 the resilience is plotted against variable diamond concentrations C , inclination angle α , and infeds f_t . The lower the ratio of diamond, the more resilient is the tool and the smaller is the resulting single grain force F_G . The expected rise of the stiffness with the inclination could not be

resilience 1/c

mm/N

C100

C50

C200

$v_c = 6.4 \text{ m/s}$
 $v_f = 500 \text{ mm/min}$
 $\alpha = 36^\circ$
D20, RVM-N

resilience 1/c

mm/N

C100

$v_c = 6.4 \text{ m/s}$
 $v_f = 500 \text{ mm/min}$
D20, RVM-N

8° 15° 26° 35° 45°

infeed f_t

mm

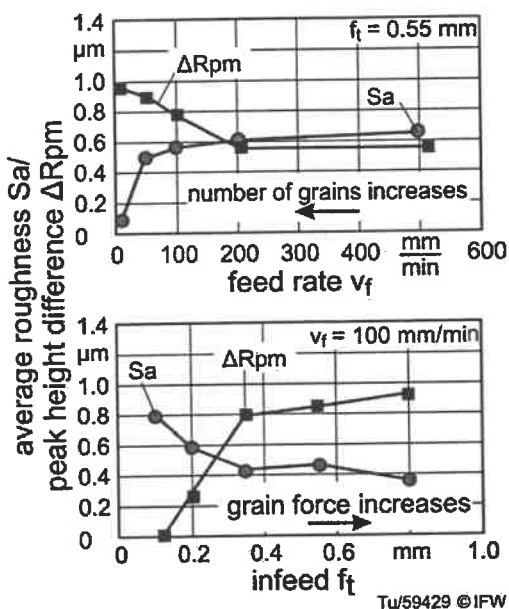
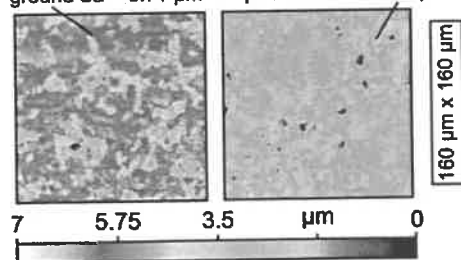
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Besides stiffness and contact area investigations, polishing experiments with variable tool specification (C , d_g , bonding) and different process parameter settings (v_c , v_f , f_i , f_p , α) have been carried out. Figure 6 shows the exemplarily influence of the feed rate v_f and the infeed f_i on the workpiece quality for a silicone

Roughness parameters:
Sa EUR 15178N (RMS of 3D surface)
Rpm DIN EN ISO 4287, ASME B46.1,
mean of all profiles, cut-off 0.08 mm

Tool:
silicone bond,
C100
 $d_g = 20 \mu\text{m}$
 $v_c = 6.4 \text{ m/s}$
 $\alpha = 45^\circ$

ground Sa = 0.71 μ m polished Sa = 0.37 μ m



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bond diamond tool with grain size D20 (15 - 25 μm) by concentration C100. This is described by the average surface roughness S_a and the reduction of roughness peaks (peak height R_p). The surfaces were measured by white light interferometry (μsurf , Nanofocus) and determined by EN15178 (S_a) and DIN EN 4287 (R_{pm}).

In Figure 6 it turns out that the roughness S_a of polished surfaces decreases the slower the tool moves (low v_t) due to more grains sliding over the surface during a specific time unit. This also results in a higher peak height (ΔR_{pm}). The situation is different with increasing infeed f_t of the tool to the surface (Figure 6, bottom). This mechanism can be described as follows. The more the tool is pressed onto the surface the higher the force which pushes the grains into the surface and the stronger the smoothing of the roughness peaks (Figure 7).

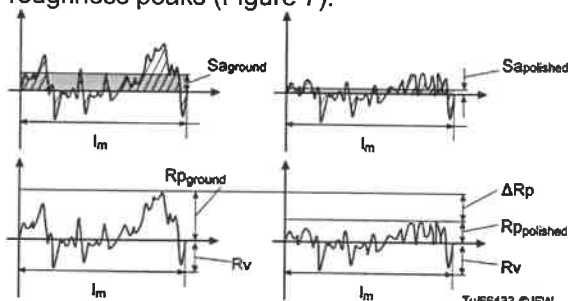


FIGURE 7. Change of roughness parameters during polishing process with resilient tools.

The results clearly indicate a relationship between the achievable roughness S_a , the number of active grains and the grain force. The roughness decreases progressively with an increasing number of diamond grains; while raising the single grain force. The influence of the grain force is 30 % lower than that of the number of grains due to very low process normal forces ($< 10 \text{ N}$).

CONCLUSION AND OUTLOOK

A theoretical model determining the active grains and the acting single grain force was developed. Thru polishing experiments, the resulting roughness has been evaluated. Further investigations will focus on the influence of the initial roughness after grinding, different bonding materials, the grain and pore size of the ceramic material, as well as the tool wear on the surface roughness to expand the process model. Prospectively, the impact of the grain force will be investigated by means of force controlled

scratch tests under real single grain loading to determine the single grain chip thickness during polishing. Finally, a complete process model to predict the surface quality after the grinding and polishing of ceramic knee implants will be generated.

ACKNOWLEDGMENT

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SUPPRESSION OF TOOL WEAR IN DIAMOND TURNING OF STEELS BY SURFACE MODIFICATION – Effect of Carburizing –

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INTRODUCTION

Severe tool wear occurs when commercial steel alloys are single-point diamond turned (Fig. 1). Consequently, suppression of tool wear in diamond turning of steels are awaited in the manufacture of molds for high-precision, complex optical components.

In previous paper, we have reported that the formation of small carbon-compound particles that are uniformly dispersed within a continuous α -ferrite matrix suppresses nose wear of diamond tools [1] (Fig. 2).

In this paper, wear suppression effect of surface modification (e.g., carburizing and nitriding) is discussed based on the results of cutting test and metallographic analysis.

MACHINABILITY OF STEEL

EXPERIMENTAL METHOD

Figure 3 shows SEM micrographs showing microstructure of quenched and tempered hot work die (JIS SKD61) steel, carburized one, and nitride one.

Figure 4 shows X-ray diffraction patterns for quenched and tempered steel, carburized one, and nitrided one.

These analyses revealed that the steels had considerably different microstructures: quenched and tempered steel consists of α -ferrite; carburized one consists of α -ferrite and carbide compounds (e.g. Fe_3C and Cr_{23}C_6); nitrided one consists of α -ferrite and nitride compounds (e.g. γ' - Fe_4N , ϵ - Fe_3C , and CrN).

Those steels were single-point diamond turned, and tool wears were measured with a scanning electron microscope.

Table 1 shows the detailed tool geometries and cutting conditions.

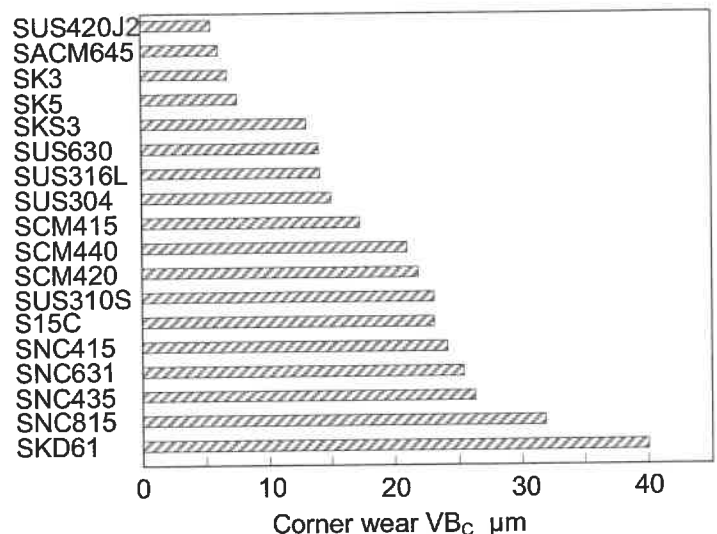


FIGURE 1. Corner wear of diamond tool when turning different types of quenched and tempered steels. $V=3.3\text{m/s}$, $f=3.1\mu\text{m/rev}$, $d=3.5\mu\text{m}$, $L=16\text{m}$.

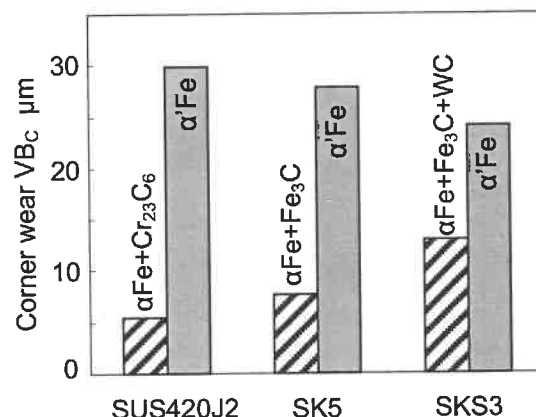


FIGURE 2. Corner wear of diamond tool when turning quenched and tempered steels (checked bars) and quenched steels (grey bars).

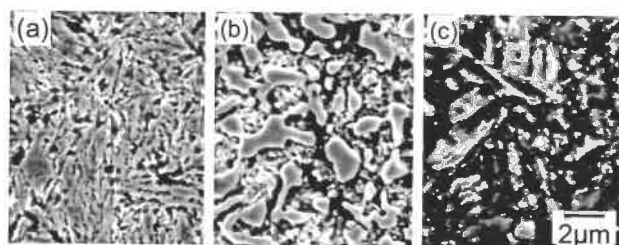


FIGURE 3. SEM Micrographs showing microstructure of (a) quenched and tempered JIS SKD61 steel, (b) carburized one, and (c) nitrided one.

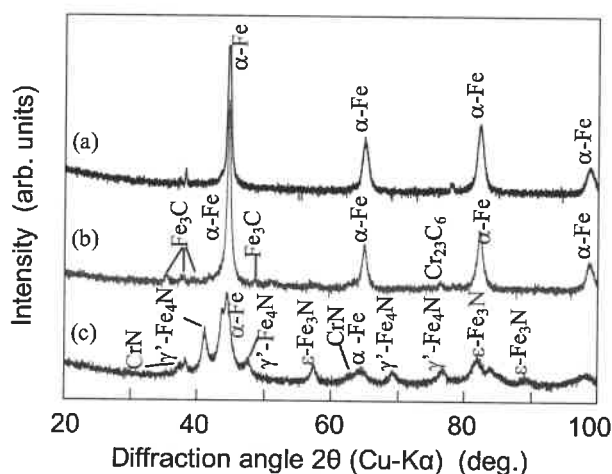


FIGURE 4. X-ray diffraction patterns for (a) quenched and tempered JIS SKD61 steel, (b) carburized one, and (c) nitrided one.

TABLE 1. Machining parameters used to assess the flank wear on diamond tools.

Cutting tool	
Material	Monocrystalline diamond Ia
Crystallographic orientation	(100) rake plane / (110) front plane
Included angle	130°
Rake angle	0°
Clearance angle	7°
Cutting conditions	
Cutting speed	0.67-3.35 m/s
Feed rate	3.1 µm/rev
Depth of cut	3.5 µm
Cutting length	16 m
Coolant	Air

RESULTS AND DISCUSSIONS

TOOL WEAR

Figure 5 shows SEM micrographs of flank of diamond tool after turning of quenched and tempered JIS SKD61 steel, carburized one, and nitrided one.

Maximum wear was observed at the tool nose, and its extent varied for surface modifications.

Figure 6 shows SEM micrographs of tool geometry after turning of quenched and tempered JIS SKD61 steel, carburized one, and nitrided one.

The diamond tool was worn out by quenched and tempered steel, on the other hand, carburized one and nitrided one did not wear out diamond tools severely

Figure 7 shows comparison of corner wear of diamond tools after turning of quenched and tempered steel, carburized one, and nitrided one in different cutting speed.

These results indicate that carbon compound precipitation by carburizing of steels suppresses nose wear of diamond tools, and that nitride formation by nitriding of steels also suppresses nose wear of diamond tools.

These results also indicate that corner wears of diamond tools differ in different cutting speed.

In general, cutting temperature increases as cutting speed increases. We compared wear rate in different cutting speed because diamond tool in turning of steel wears thermodynamically [2]. Figure 8 shows comparison of wear rate as a function of inverse of cutting speed. Wear rates increases exponentially as the cutting speed increases. Slopes of approximated lines are in the same order.

Wear rate when turning carburized steel was equivalent to 1/4 of quenched and tempered one, and that of nitrided one was about 1/30.

SURFACE INTEGRITY

Figure 9 shows finished surface of quenched and tempered JIS SKD61 steel, carburized one, and nitrided one.

Comparison of surface roughness in different cutting speed is shown in Figure 10.

These results indicate that diamond tools generate worse surface roughness than theoretical roughness (i.e. 0.74µm) when turning steels though their edges are worn out.

To clarify this fact, we calculated roughness factor of each case. Figure 11 shows comparison of roughness factors of quenched and tempered JIS SKD61 steel, carburized one, and nitrided one.

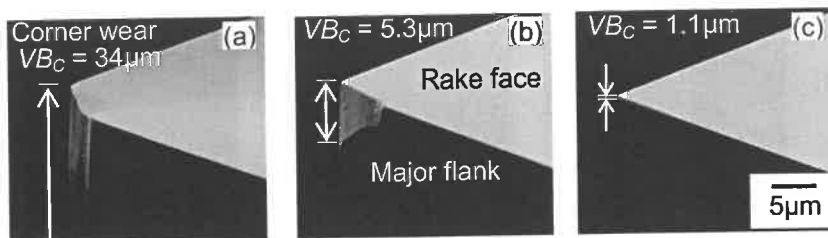


FIGURE 5. SEM micrographs of flank of diamond tools after turning of (a) quenched and tempered JIS SKD61 steel, (b) carburized one, and (c) nitrided one. $V=3.3\text{m/s}$, $f=3.1\mu\text{m/rev}$, $d=3.5\mu\text{m}$, $L=16\text{m}$.

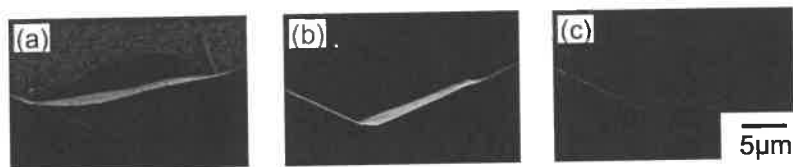


FIGURE 6. SEM micrographs of tool geometry after turning of (a) quenched and tempered JIS SKD61 steel, (b) carburized one, and (c) nitrided one. Dotted lines indicate tool geometry before turning.

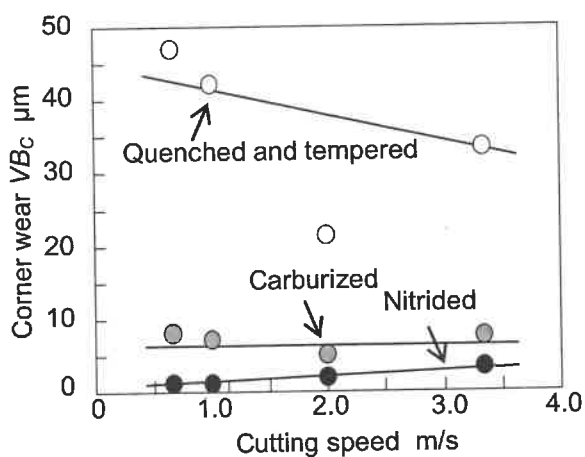


FIGURE 7. Comparison of corner wear of diamond tool versus cutting speed.

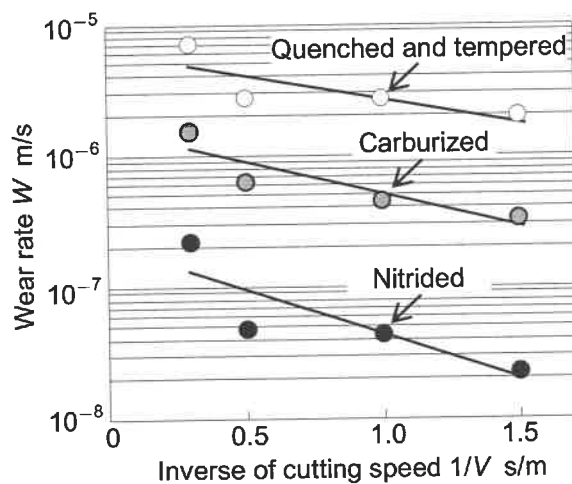


FIGURE 8. Comparison of wear rate as a function of inverse of cutting speed.

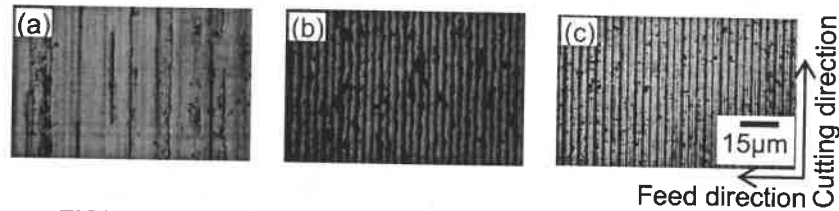


FIGURE 9. Finished surface of (a) quenched and tempered JIS SKD61 steel, (b) carburized one, and (c) nitrided one.

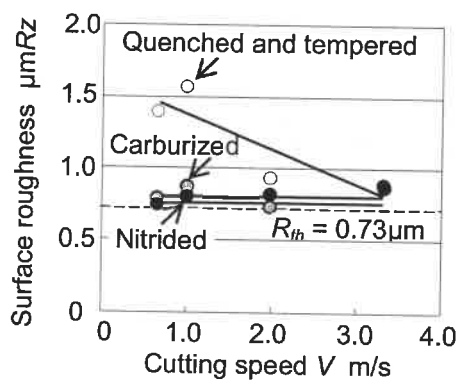


FIGURE 10. Comparison of surface roughness as a function of cutting speed.

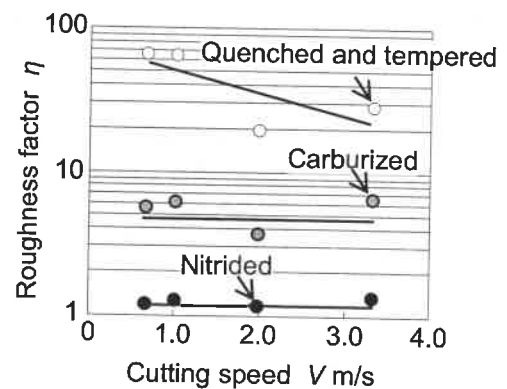


FIGURE 11. Comparison of roughness factor of quenched and tempered JIS SKD61 steel, carburized one, and nitrided one.

Roughness factor when turning quenched and tempered steel was more than 10, whereas that of carburized steel was equivalent to 5, and that of nitrided one was almost equivalent to 1. These results indicate that carburizing and nitriding improves surface integrity.

CONCLUSIONS

- Carbon compound precipitation by carburizing of steels suppresses nose wear of diamond tools.
- Nitride formation by nitriding of steels suppresses nose wear of diamond tools.

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EFFECT OF VACUUM ON WIRE-SAWING AS PRELIMINARY EXPERIMENT TO LUNAR EXPLORATION

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INTRODUCTION

Bullet type samplers are sometimes used to obtain samples from asteroids in sample-return missions [1]. On the other hand, in-situ analysis is demanded for the investigation of more amounts of rock samples. Many drilling or coring devices have been developed [2-4]. Slicing the rock sample is preferable to observe its interior rather than shaving [5, 6]. Because coolant cannot use in vacuum environment, tool life will be shorten. A saw wire may keep cutting performance by supplying successive cutting edges. This paper describes some machining performance with a prototype of a wire-sawing machine.

PROTOTYPE OF WIRE-SAWING MACHINE

Figure 1 and Table 1 show an appearance and specifications of a prototype. It was installed in a vacuum chamber and was driven with two 50-W AC servo motors mounted on the outside of the chamber with rotational feedthroughs. Figure 2 shows a wire path. A feed length was measured

with a rotary encoder. While a saw wire was reciprocated, one motor wound at a constant velocity and another applied the tension by a PID control. The maximum speed of the saw wire was 2 m/s by the limitation of the motor torque. A workpiece was mounted on an end of a lever and a cutting load, namely normal force, was applied with a constant-force spring. An edge of each grain sticks on electroplated nickel bond on a bare saw wire, and nickel bond completely covers grains on a Ni-coated one.

CUTTING EXPERIMENTS

Conditions of Wire-sawing

The cutting performance in vacuum was compared with that in air. Table 2 shows machining conditions. A saw wire with 30-40- μ m diamond grits was used. A 14-m the saw wire was used in each experiment (10 m for a constant speed). A terrestrial basalt block, which has a similar structure and strength to a lunar mare basalt rock, with dimensions of 10 mm $\times$ 15 mm $\times$ 15 mm was used for a workpiece.

When a workpiece was cut with an unused Ni-

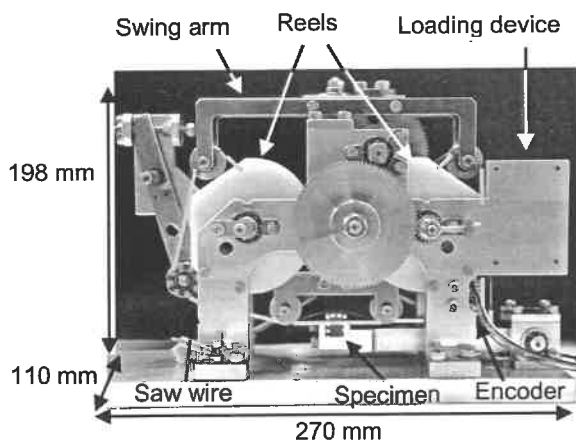


Figure 1. Wire-sawing machine.

Table 1. Specifications of wire-sawing machine.

Dimensions	270 $\times$ 110 $\times$ 198 mm	
Weight	6.0 kg	
Maximum speed	2.0 m/s	
Structural materials	ASTM-5052, ASTM-S30400, Polyacetal	
Saw wire	Abrasive	30-40 μ m diamond
	Bond	Electroplated nickel
	Diameter	0.28 mm
	Maximum length	600 m

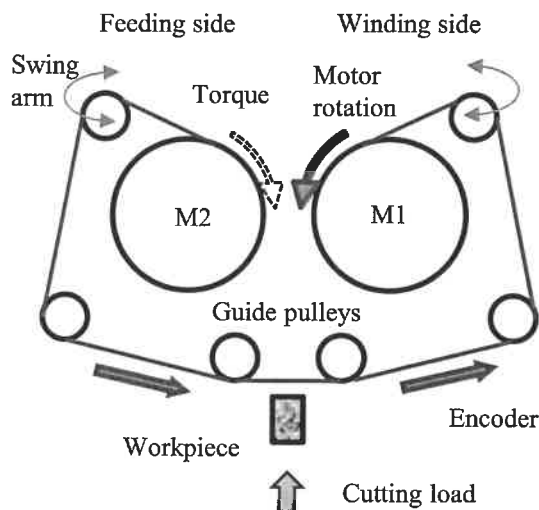


Figure 2. Wire path under tension and velocity control.

Table 2. Machining conditions.

Wire speed	0.1-1.0 m/s
Wire length	14 m
Cutting load	0.8, 1.6 N
Wire tension	2 N
Number of reciprocation	15
Pressure	10^{-5} , 10^{-3} , 10^{-5} Pa
Workpiece	Basalt mined in Japan

coated saw wire, it was seldom cut and the machined groove was glazed. Figure 3 shows an expanded view of the Ni-coated saw wire used at a pressure of 10^{-3} Pa and results of energy dispersive X-ray spectroscopy (EDS) analysis. Main components of basalt, Si and Al, were observed near a grain. Debris was adhered there by electrification. Figure 4 shows a machined groove and results of EDS analysis. The bond material, Ni, was observed there. Because basalt is harder than electroplated nickel, the bond was scraped with pores and adhered there. The diamond grains slipped on the nickel adhesion. The Ni-coated saw wire was dressed with other basalt samples by reciprocating 10 times before cutting tests.

Influence of Machining Conditions

Wire-sawing was tried 3 times in vacuum and twice in air because the variation of machined depth at each condition was small in air. In order to hold down the power consumption of the motors and due to the torque capacity, the saw wire was fed below 1 m/s. The Ni-coated saw

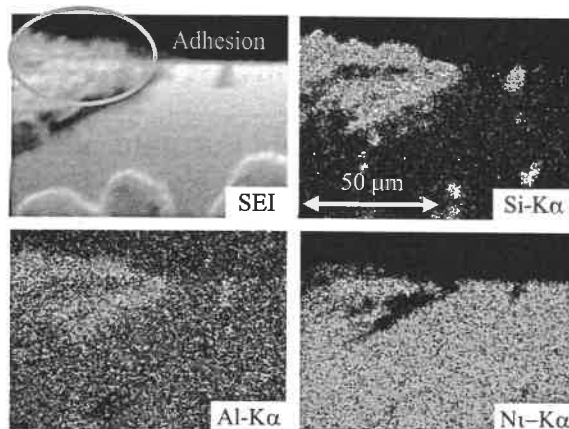


Figure 3. Surface of saw wire after cutting in vacuum.

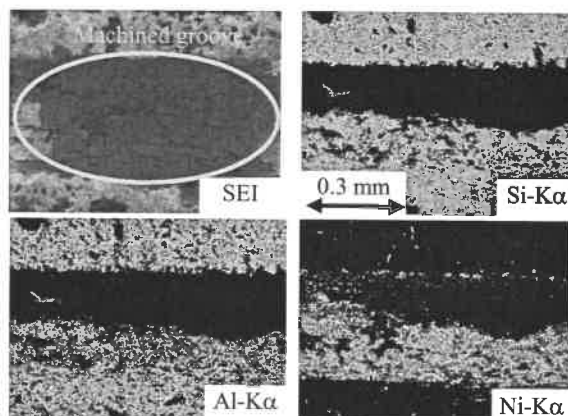


Figure 4. Surface of machined groove.

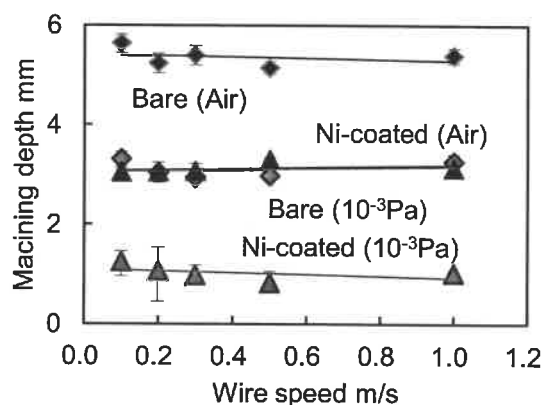


Figure 5. Machining depth vs. wire speed.

wire was mainly used in the following experiments.

Figure 5 shows the relationship between the machined depth and wire feeding speed. The

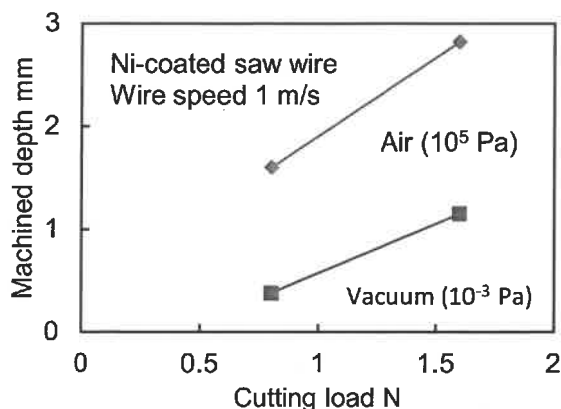


Figure 6. Machined depth vs. cutting load.

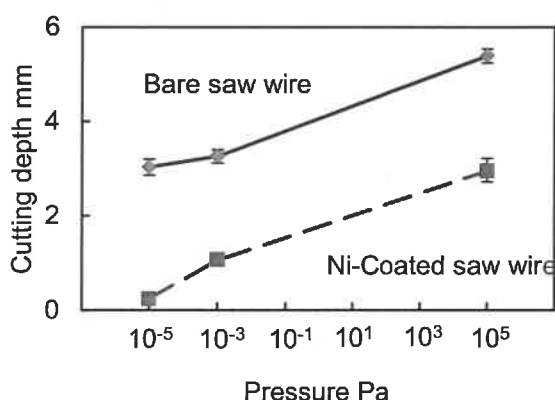


Figure 7. Machined depth vs. vacuum pressure.

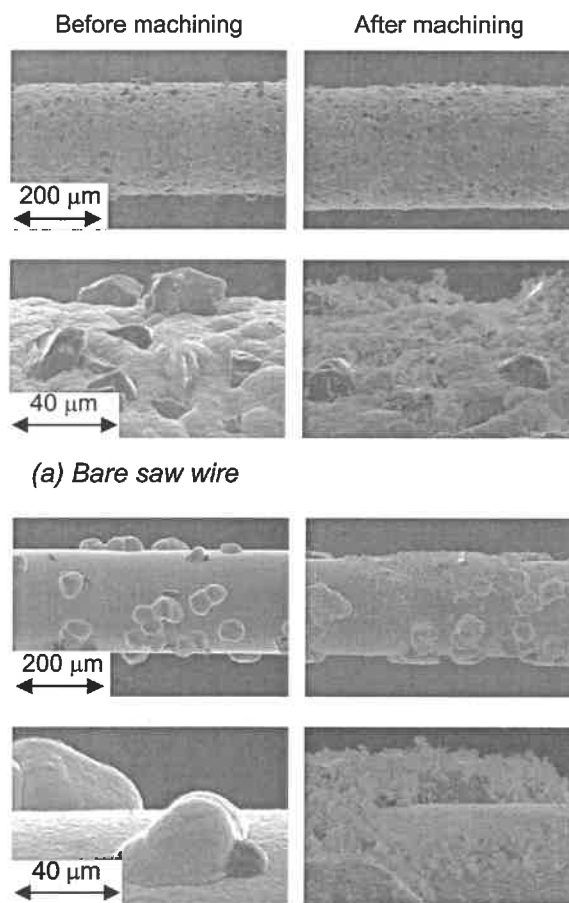
cutting load was set to 1.6 N. The wire feeding speed did not affect the cutting depth below 1 m/s.

Figure 6 shows the relationship between the machined depth and cutting load. The wire feeding speed was set to 1 m/s. With an increase of cutting load, the cutting depth was increased.

These trends were the same both in air and vacuum.

Effect of Initial Surface of Saw Wire

The cutting performance of the Ni-coated saw wire was compared with the bare one at various vacuum pressures. Figure 7 shows the relationship between cutting depth and pressure. The wire feeding speed and cutting load were set to 1 m/s and 1.6 N, respectively. The cutting depth was decreased in vacuum. The cutting



(b) Ni-coated saw wire

Figure 8. Saw wires used in vacuum.

depth with the bare saw wire was larger than that with the Ni-coated one.

Figure 8 shows appearances of the saw wires before and after machining. The photographs do not show the same position before and after machining. Most grains remained on the nickel bond on the bare saw wire. Debris was not adhered on some grains. The grains completely remained on the Ni-coated saw wire because of larger gripping force than the bare one. Worn nickel bond always spread around the machined area.

Observation of Machined Surface

Machined surfaces of a rock sample were observed. The machining conditions were the same as Table 2. The vacuum pressure was $1-3 \times 10^{-3}$ Pa during machining. The number of reciprocating motions of the bare and Ni-coated saw wires was 40 and 90 times, respectively. A $3 \times 4 \times 10$ -mm cuboid was cut off from the sample.

Table 3 Observed positions.

		Bare	Ni-coated
Machined depth mm		4.0	3.0
Observed position from top mm	Entrance	0.5	0.5
	Middle	2.0	1.5
	Bottom	3.5	2.5

Table 3 shows observed position of the machined specimen. Three areas were observed: 0.5 mm from the edges and middle.

Figure 9 shows nickel component on the machined surfaces of the rock samples analyzed by EDS. Bright areas contain nickel. Although nickel was observed at all area cut with the Ni-coated saw wire, it was observed only the bottom area cut with the bare one. With the increase of wear of the grains, the nickel bond of the bare saw wire was scratched with the rock. Therefore, the bare saw wire is better than the Ni-coated one for cutting rocks in vacuum.

The surface roughness was 3.8 in orthogonal direction to the wire feeding and 1.1 μmRa in parallel. They are smooth enough to observe the structure and components of the rock samples.

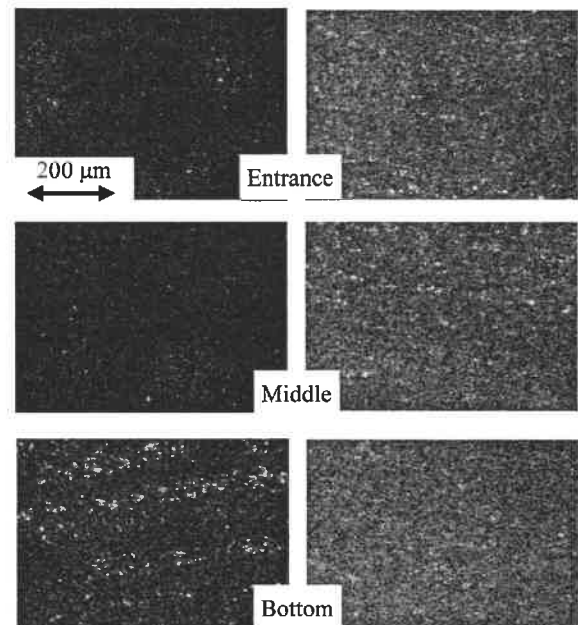
CONCLUSIONS

Rock samples were cut with saw wires in air and vacuum in this paper. Conclusions can be drawn as follows.

- (1) With the decrease of the vacuum pressure, the machining depth was decreased. The wire feeding speed did not affected machining depth below 1 m/s.
- (2) The nickel bond was adhered on the rocks during cutting and grains slipped on the nickel layer. It caused the decrease of the machined depth.
- (3) The bare saw wire was preferable for machining rocks in vacuum.

The wear rate of the grains will be investigated for a future work.

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(a) Bare saw wire (b) Ni-coated saw wire

Figure 9. Nickel component on machined rock surface analyzed by EDS.

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Investigation into Surface Characteristics of Cylindrical Components Machined using Wire Electrical Discharge Turning (WEDT) Process

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INTRODUCTION

Wire-Electrical Discharge Machining (Wire-EDM or WEDM) is one of the important non-traditional machining processes, which is used for machining of difficult-to-machine materials and intricate profiles. Turning with WEDM is one of the emerging areas, developed to generate cylindrical forms on hard and difficult to machine materials by adding a rotary axis to WEDM. Several researchers have contributed for turning of components using wire electrical discharge machining. Turning of small diameter pins of size 5 μ m and micro spindles using WEDM, which can be used for making products where the roughness and dimensional accuracy can be of order of ten nanometers [1]. Machining parameters are varied to explore the possible surface finish and roundness by cylindrical WEDM process [2]. An attempt has been made to study the effect of machining parameters on material removal rate, surface roughness (Ra) and roundness of parts machined using cylindrical Wire Electrical Discharge Turning (WEDT) process [3]. However, it is observed from the literature that not much work has been carried out in analyzing the surface characteristics of components machined using wire electrical discharge turning process.

In this paper the influence of input parameters such as pulse off time, servo feed and spindle rotation will be analyzed from roundness, surface roughness and energy of machined components, which are taken as the output parameters. EN 24 high tensile steel is used as the work piece material. Generally EN 24 steels are used for heavy duty axle shafts, pinions, torsion bars, etc.

EXPERIMENTAL DETAILS

Experiments are conducted on Electronica ECOCUT machine with four controllable axes.

Brass wire of diameter 0.25mm is used as the electrode and distilled water as the dielectric medium. An external rotary setup developed by [4] is used for giving rotary motion for the work piece. The experimental setup for conducting experiments is shown in Figure 1. The pulse off time can be varied from 9 to 42 μ s, servo feed from level 1 to 10, and spindle rotation from 4 to 100 rpm. Among these parameters the pulse off time is varied at 15 μ s, 28 μ s and 40 μ s. Servo feed level is varied at 3, 6 and 8. Spindle rotation is varied at 30, 60 and 90rpm. Spark gap and wire feed are maintained constant at 40 μ m and 3 m/min respectively. The parameters used for conducting the experiments are given in Table1.

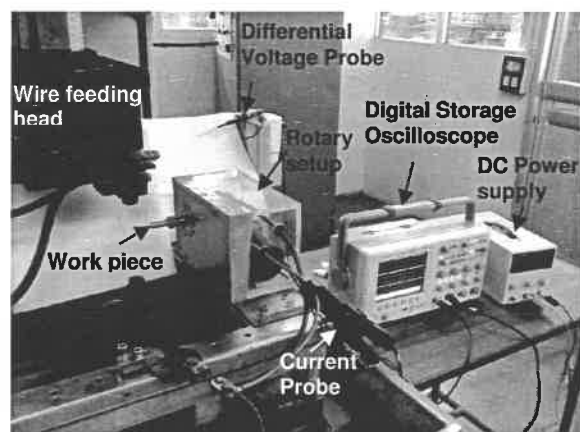


FIGURE 1. Experimental setup of WEDT

TABLE 1. Machine setting parameters

Pulse off time (μ s)	Servo feed (Level 1-10)	Spindle rotation (rpm)
15	3	30
28	6	60
40	8	90

The roundness is measured using Mahr MMQ 200 and Figure 2 shows the typical roundness profile before and after machining. Surface roughness is measured using Mahr Perthometer M2 and Figure 3 shows the typical roughness profile before and after machining.

Energy conservation has become a crucial subject and has generated interest amongst production engineers in considering the fundamental issues directly affecting the choice of production method to be used in manufacturing. The energy (E) is calculated from power pulse train obtained using digital storage oscilloscope for time duration of 1 second. When $P(t)$ is the power in watts, t_1 and t_2 are the time intervals, the energy is given by the equation 1

$$E = \int_{t_1}^{t_2} P(t) dt \quad (1)$$

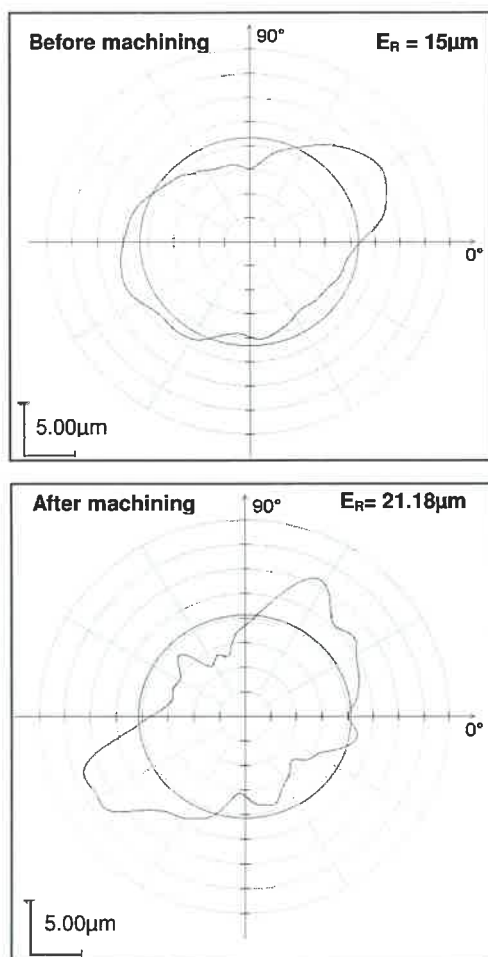


FIGURE 2 Roundness profiles of specimens

RESULTS AND DISCUSSION

Effect of machining parameters on surface roughness

The surface roughness is characterized using the arithmetic mean (R_a). The roughness is measured at three different locations on the specimen and their average is given in Table 2.

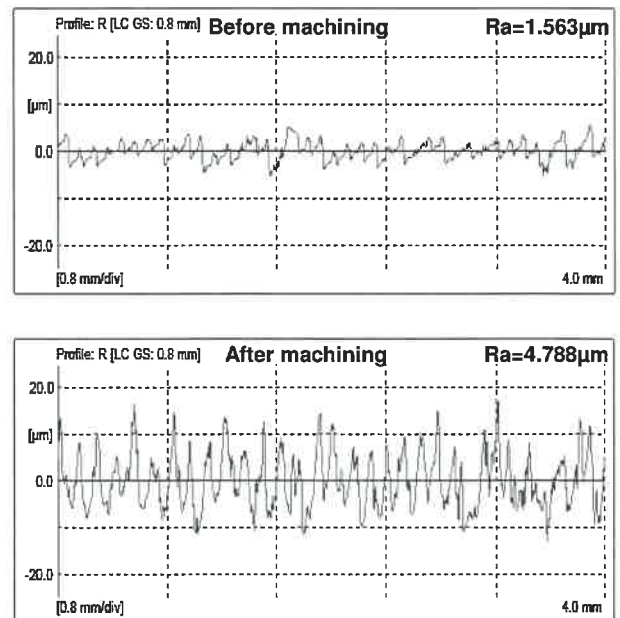


FIGURE 3 Roughness profiles of specimen no.9

For fixed pulse off time varying servo feed level and spindle rotation, roughness value increases. From Table 2 it is seen that, when servo feed level is varied from 3 to 8 surface roughness value increases. At lower values of servo feed (level 3) the gap width is larger results in higher average gap voltage. Larger the gap width lesser the debris will be formed, which results in formation of normal pulse that leads to reduced surface roughness.

Figure 4 shows the SEM image of specimen machined with pulse off time $40 \mu s$, servo feed level 3 and spindle rotation 60rpm at magnification of 250X and 1000X. From the image we can observe the crater formation on machined surface is even due to normal pulse. When the servo feed level is increased to 6, the gap width reduces results in lower average gap voltage. When gap width is small the ignition delay time is reduced which results in more number of discharges for a given time. As the number of discharges increases more amount of

TABLE 2 Experimental results for surface roughness, roundness and energy of machined components

Specimen No.	Pulse off time (μ s)	Servo Feed Level (1-10)	Spindle rotation (rpm)	Surface Roughness (R_a) (μ m)		Roundness (E_R) (μ m)		Energy (J)
				Before machining	After machining	Before machining	After machining	
1	15	3	30	1.660	2.505	7.09	8.76	0.02530
2	15	6	60	1.818	4.062	12.16	21.03	0.00130
3	15	8	90	1.631	5.277	8.70	21.87	0.00048
4	28	3	90	1.750	2.819	7.03	15.60	0.01960
5	28	6	30	1.768	5.297	9.60	21.80	0.00150
6	28	8	60	1.600	7.677	9.65	24.80	0.00092
7	40	3	60	1.786	2.289	12.33	7.19	0.01600
8	40	6	90	1.988	4.201	7.84	21.97	0.01230
9	40	8	30	1.540	4.956	11.50	23.77	0.00900

debris will be formed, this results in arc pulse formation. The formation of arc pulse will increase the crater size on the machined surface, which increases the surface roughness. At higher values of servo feed (level 8) gap width reduces further results in reduced ignition delay and more debris to be formed in the gap. Increase in debris concentration leads to discharge localization (multiple sparking). If the discharge is localized, the distribution of gap width becomes uneven, resulting in localized crater formation which increases surface roughness. Figure 5 shows the SEM images of specimen machined with Pulse off time 28μ s, servo feed level 8 and spindle rotation 60rpm at magnification of 250X and 1000X. From the image we can observe the localized crater formation. Figure 3 shows the roughness profile of specimen machined with Pulse off time 40μ s, Servo feed level 8 and Spindle rotation 30rpm. As the machining continues more debris will be formed in the gap that leads to formation of short circuit pulses. When spindle rotation is increased from 30 to 90 rpm the circumferential length of the material crossing the sparking zone increases, which results in arc pulse formation that increase the surface roughness.

Effect of machining parameters on roundness error

Roundness value is measured at three different locations on the specimen and their average is given in Table 2. From Table 2, it is seen that roundness value increases with increase in servo feed level. When rotational speed is varied by keeping servo feed at level 3, circumferential

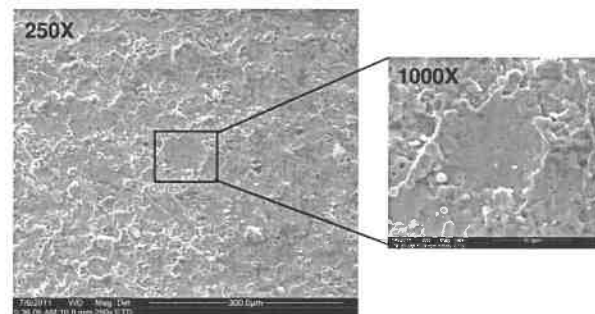


FIGURE 4 SEM images of specimen machined with Pulse off time 40μ s, Servo feed level 3 and Spindle rotation 60rpm.

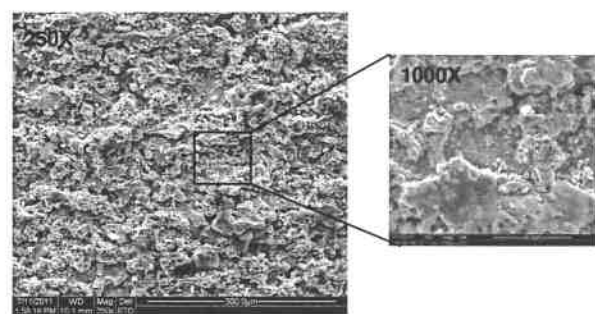


FIGURE 5 SEM images of specimen machined with Pulse off time 28μ s, Servo feed level 8 and Spindle rotation 60rpm.

length of the material crossing the sparking zone varies. As the gap width is larger, ignition delay will be more and less number of discharges occurs. This results in lower roundness values. As the rotational speed varied at servo feed level 6, the gap width got reduced which leads to reduced ignition delay. More number of

discharges will occur, that results in increased roundness value. When the rotational speed is varied at servo feed level 8, the gap width reduced further. Reduction in gap leads to accumulation of debris in the sparking zone, which in turn results in localized sparking as shown in Figure 6. Localized sparking results in increased crater size, which in turn, results in increased roundness error as seen in Figure 2.

Effect of machining parameters on Energy

In this work the energy is also found to study the effect of machining parameters. For fixed pulse off time varying the servo feed level and spindle rotation, energy shows a decreasing trend. At lower values of servo feed (level 3) the gap width is larger results in larger ignition delay. The number discharges per unit time will be less and the amount of debris formed is also small. The energy is effectively used for even crater formation. So higher will be the energy output. When servo feed at level 6, gap width is reduced which results in increased discharges per unit time. As the number of discharges increase, arc pulse will be formed. The combined normal and arc pulse will increase the debris concentration, which results in short circuit pulse to occur momentarily. Formation of short circuit pulses will reduces the energy output. At servo feed level 8, the gap width is so small that bridging of gap with debris occurs in short time. Bridging of gap with debris leads to short-circuit current flow through it, which reduces the energy output from the sparking zone.

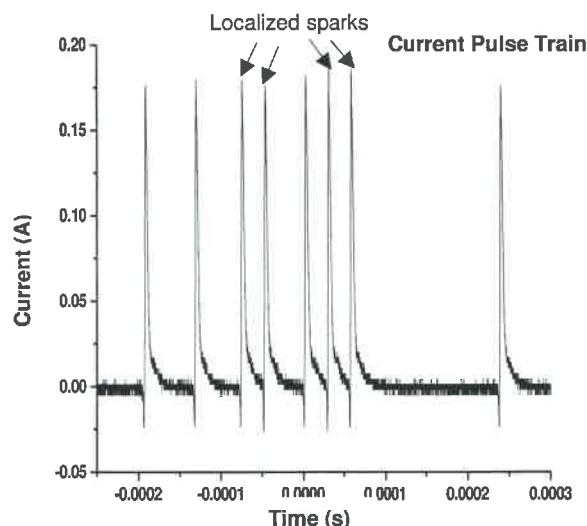


FIGURE 6 Localization of current pulses obtained for Pulse off time $28\mu\text{s}$, Servo feed level 8 and Spindle rotation 60rpm.

CONCLUSION

The effect of pulse off time, servo feed and spindle rotation on surface roughness and roundness of machined components have been analysed. It is found that for fixed pulse off time, surface roughness increases in varying servo feed level and spindle rotation. Similarly, roundness value is also increases with in varying servo feed level and spindle rotation for fixed pulse off time. This is due to the formation of arc pulses and localized sparks which result in increased crater size on the machined surface.

Energy is found to be decreases when varying servo feed level and spindle rotation for fixed pulse off time. This is due to formation of more debris in sparking gap, that results in bridging of gap with debris which makes short-circuit current flow through it. The effect of these machining parameters can be studied to get the optimum energy consumption during machining for achieving better MRR and surface property of the machined parts. This work can be applied for on-line monitoring and optimization of power consumption during WEDT process.

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MANUFACTURING OF HIGH QUALITY SURFACES ON A FINE GRINDING MACHINE WITH LAPPING KINEMATICS

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INTRODUCTION

In order to achieve high quality plane surfaces with very good evenness and low roughness, fine grinding with lapping kinematics becomes more important, and is nowadays state of the art. In comparison to conventional lapping using slurry, fine grinding with lapping kinematics using bonded wheels has several advantages. This process is characterized by low surface zone damage. Thus, the process time of the following process, mainly polishing, can be decreased and the costs are reduced. Nevertheless fine grinding with lapping kinematics still has several disadvantages. The wear of the wheel is not constant over the diameter, thus, the wheel does not remain flat [1-4]. The Institute of Machine Tools and Production Technology established a novel machine concept, which uses foil tools instead of grinding wheels, presented at ASPE 2009 conference [5]. Thus, the tool part, determining the surface roughness is separated from the on determining the surface form. Since the foil tool is continuously driven over the plate, a constant machining behavior is achieved. Thus, a very good evenness can be ensured during the total process time without the necessity of time consuming dressing of the wheel. That way the disadvantages of conventional fine grinding with lapping kinematics could be solved while the advantages were kept.

However, process kinematics highly affects the machining process regarding material removal depth and evenness. Chips can lead to scratches on the machined surface and reduce the achievable surface quality. Thus, the influence of process kinematics and filtration was investigated during finishing of hardened steel Ck45.

INFLUENCE OF FILTRATION

Finishing with lapping kinematics usually takes place within three process steps. During the first step, a comparatively rough abrasive foil tool with grain size 30 μm is used to improve the evenness. In the second step an abrasive foil tool with grain size 15 μm is used for further material removal to remove scratches from the surface. The last process step is conducted using an abrasive foil tool with grain size 9 μm to achieve the required surface roughness. A 3 % emulsion was used for all experiments.

All process steps are operated on the same machine. Thus, chips from previous machining processes remain in the coolant. Therefore, the usage of filtration on machining results was investigated. The difficulty regarding filtration is the small grain and chip size. Hence, the mesh size of the filter should be very small. However, a small mesh size may also remove additives from the coolant, since the particle size of emulsions is in the range of 1 to 10 μm . Thus, a compromise regarding mesh size had to be found. Previous experiments pointed out, that a 10 μm mesh size is not sufficient for filtering since chips and grains from previous experiments remain in the coolant. Thus, a cartridge filter 3M Cuno type 744 B30 with mesh size 5 μm was used.

During each test run four cutting speeds (0.2, 0.4, 0.6 and 0.8 m/s) were investigated. One test lasted 8 minutes. Prior to the test and in steps of 2 minutes the test was interrupted to measure material removal rate, evenness and roughness. Furthermore, 100 ml of coolant were taken from the tank to judge the amount of chips remaining in the coolant. First of all, six test runs were realized using the cartridge filter, followed by six test runs without any filtration. In order to avoid faults in the grinding results due to filtering additives out of the coolant, the coolant was not changed after the cartridge filter has been

removed. Prior to machining, the workpieces were roughened for two minutes. Thus, the evenness has been between $f_e = 4.2 \mu\text{m}$ and $13.4 \mu\text{m}$, and the roughness was between $R_a = 0.1 \mu\text{m}$ and $0.22 \mu\text{m}$, $R_z = 0.5 \mu\text{m}$ and $1.8 \mu\text{m}$ and $R_{\text{max}} = 1 \mu\text{m}$ and $2.5 \mu\text{m}$.

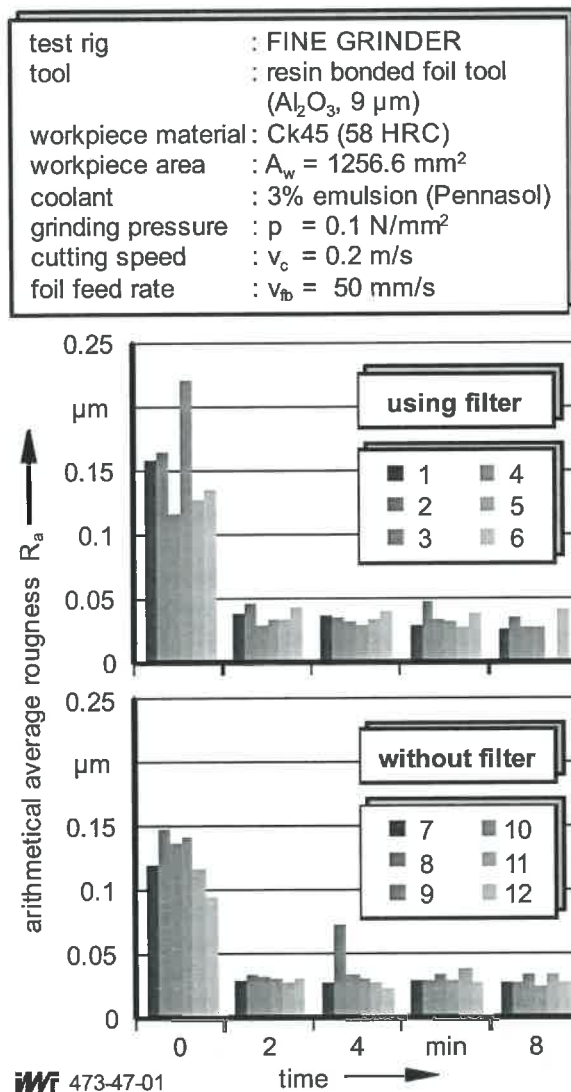


FIGURE 1. Arithmetical average roughness

It was found, that the usage of a cartridge filter is highly influencing the material removal rates. More material has been removed when using the filter, due to the fact, that fewer chips remained in the coolant aiding clogging of the foil tool. Thus, the foil feed rate should be increased to achieve the same material removal rate without using the filter as with using the filter. The influence of the cartridge filter on material removal rate was even higher than that

of the cutting speed. The repeatability of the tests was very good regardless usage of the cartridge filter.

As expected the evenness f_e was about $3 \mu\text{m}$ over the workpiece diameter of 42 mm regardless using a cartridge filter or not. The main improvement of evenness was achieved within the first 4 minutes of machining, since this time was required to remove the evenness fault of the roughing process. Furthermore, the differences in evenness using various cutting speeds were only marginal and thus, can be neglected.

Mainly, there is a good repeatability in roughness when not using the cartridge filter. Nevertheless, in some cases higher roughness values appear (figure 1). This is due to the fact, that burrs are formed during the fine grinding process at the edge of the workpiece. At some point, the burr is removed from the workpiece and carried away by the coolant. If a filter is used, this burr is removed from the coolant. Without a filter, the burr is carried into the grinding zone once again leading to scratches on the workpiece surface and thus, worsens the surface roughness.

Without using a filter the average roughness values of different cutting speeds were scattering a lot not allowing determining an influence of cutting speed on surface roughness. Nevertheless using the cartridge filter, it can be seen, that the surface roughness decreases with increasing cutting speed.

INFLUENCE OF KINEMATICS

In order to investigate the influence of the kinematics on the machining results, first of all a kinematics model of the fine grinding machine was built using Matlab. This model allows calculating the change of cutting speed for one point of the workpiece over time and furthermore, the change of average cutting speeds over the workpiece diameter. Besides that, the influence of the foil feed rate on foil usage can be investigated. Using this model, different pairs of table and workpiece rotation speeds were chosen to examine the influence of cycle time, cycle length, kinematic factor, as well as average cutting speed on machining results. The ration between table and workpiece rotation is given by the kinematic factor K , which is parameter describing the cycloidal pathways of

the workpiece (figure 2). The time until one workpiece point is back on the original point is known as the cycle time and the cycle length describes the length of this pathway. For a kinematik factor unequal to 0, the cutting speed is varying of time and over the workpiece diameter [6]. A kinematic factor of 0 means, that each workpiece point is describing a cycle. In all other cases, each workpiece point is describing an epicycloidal or hpocycloidal pathway. Therefore, pairs of rotation speeds presumed to achieve very good surface quality and those presumed to be inappropriate were compared.

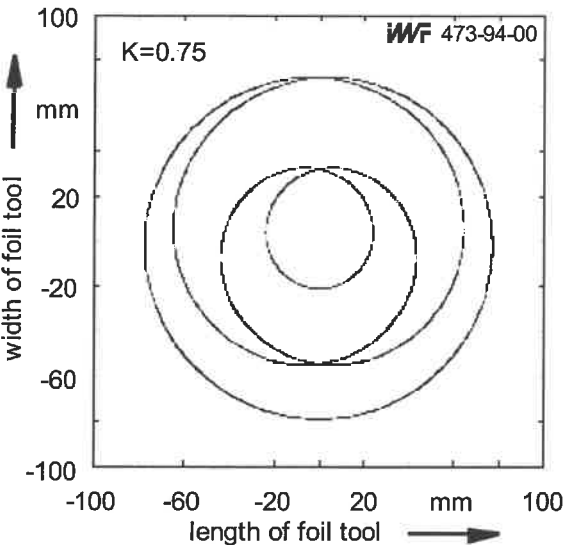


FIGURE 2. Pathway of one workpiece point using a kinematic factor of $K = 0.75$

The experiments pointed out, that cycle length is highly influencing the surface roughness but does not have the suspected influence on the material removal rate and evenness. Nevertheless, the cycle time is affecting all process output parameters. Though, in some cases the affect is only marginal and thus, might be generated by other parameters e.g. the kinematic factor. Regarding the kinematic factor, there are contrary affects for synchronous and asynchronous rotation of workpiece and table which are further investigated by now. Increasing the cutting speed leads to higher material removal rates accompanied with reduced material removal depth per meter of cycle length.

test rig	: FINE GRINDER
tool	: resin bonded foil tool (Al_2O_3 , 9 μm)
workpiece material	: Ck45 (58 HRC)
workpiece area	: $A_w = 1256.6 \text{ mm}^2$
coolant	: 3% emulsion (Pennasol)
grinding pressure	: $p = 0.1 \text{ N/mm}^2$
foil feed rate	: $v_{fb} = 50 \text{ mm/s}$

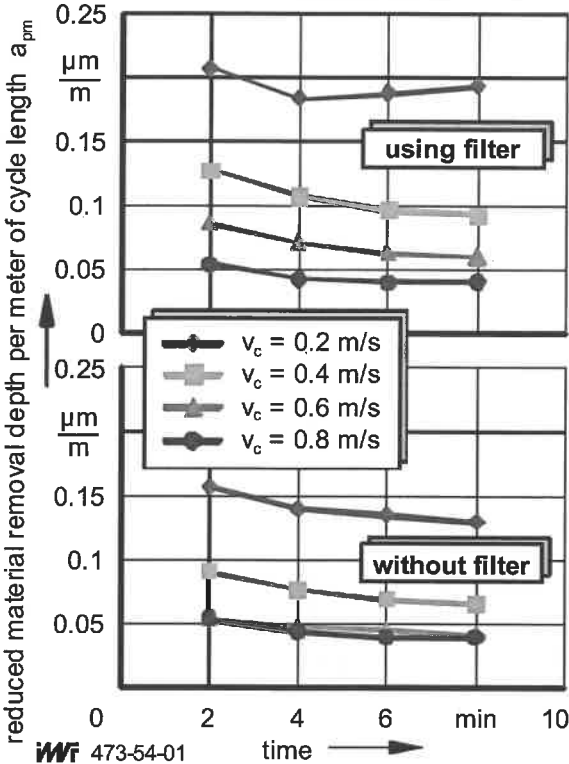


FIGURE 3. Influence of cutting speed regarding reduced material removal depth per meter of cycle length

Furthermore, a Matlab model was used to calculate the cycle time and length in order to evaluate the influence of cutting speed on reduced material removal depth per meter of cycle length (figure 3). It can be seen that the material removal depth per meter of cycle length is reduced with rising cutting speed. This means, that the grains do not enter into the workpiece as deep as with lower cutting rates, leading to smaller surface roughness. Since the material removal rate was higher using the cartridge filter, the roughness was slightly worse. The experiments also pointed out, that there is a foil tool dependant lowest possible surface roughness. Using the 15 μm foil tool, this roughness was reached at $R_a = 0.022 \mu m$.

CONCLUSION

The machine concept "Fine Grinder" combines the advantages of both, lapping and conventional fine grinding with lapping kinematics. High quality surfaces can be produced within a comparatively short time of only a few minutes depending on the input surface roughness (figure 4). Due to the separation of shape and roughness determining components high evenness can be produced. Though, kinematics is highly affecting the surface quality. Thus, a kinematics model was built in order to help evaluating the process output parameters. Experiments pointed out that cycle time, cutting speed and kinematics factor had the highest influence. Though, due to the interrelationship of those parameters further experiments are conducted by now to find proper process parameters for finishing Ck45.

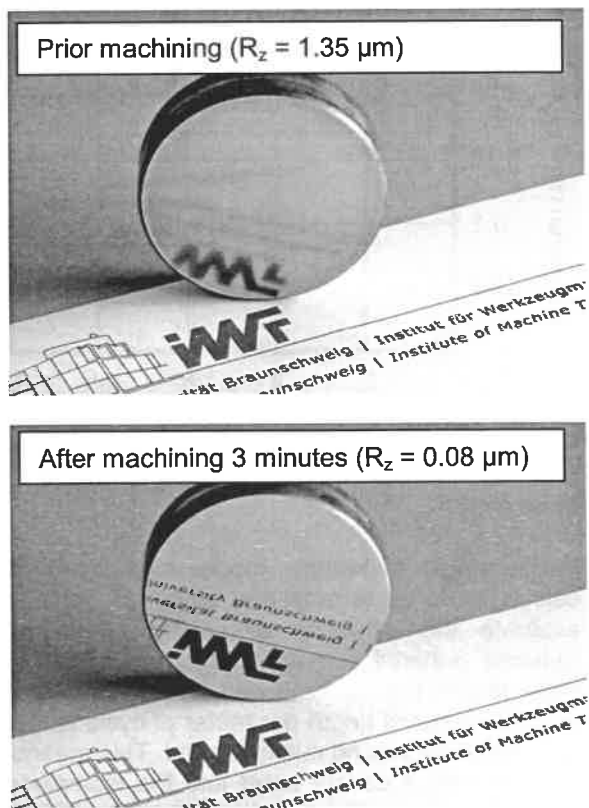


FIGURE 4. Machining results

Experiments on the usage of filters pointed out that a mesh size of 5 µm is required to remove the small chips out of the coolant. Thus, a higher, constant material removal rate could be achieved. Furthermore, burrs produced during fine grinding and removed from the workpiece

were filtered out of the coolant. Thus, scratches on the surface after finishing could be impeded. The scattering of roughness values was decreased when using the cartridge filter, allowing evaluating the influence of cutting speed on surface roughness. The surface roughness decreased with rising cutting speed until a foil tool dependent roughness limit was reached. The differences in surface roughness were analog to the reduced material removal depth per meter of cycle length.

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SUPERFINISHING OF BEARING RINGS USING ELASTIC BONDED GRINDING WHEELS

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INTRODUCTION

The running behavior of ball bearings is highly affected by the surface topography of the bearing rings. Radial deviations in form of waviness lead to vibrations and thus, noise disturbance. The surface roughness is highly affecting the life time of a ball bearing. The parameters of the Abbott-curve are describing the tribological behavior of the surface. The core roughness depth R_k and reduced peak height R_{pk} are supposed to be small. These parameters characterize the roughness peaks of the topography which are removed during the first application hours of the ball bearing, leading to particles which further enhance the ball bearing wear. The reduced valley height R_{vk} should have a comparatively higher value, because the valleys are keeping the lubricant in the friction zone.

Today, bearing rings are made of hardened bearing steel 100Cr6. The last two process steps are grinding and superfinishing. The superfinishing process is conducted at a separate machine using honing stones [1, 2]. This requires a new clamping of the bearing ring, reducing the form accuracy and increasing the non-productive time in production.

This paper deals with a novel superfinishing process using elastic bonded grinding wheels instead of honing stones. Thus, finishing and superfinishing can be performed on one single machine leading to higher form accuracy and lower non-productive time. The influence of the grinding wheel specification, process parameters and type of coolant on finishing results were investigated and presented in this paper. The tests were performed on outer bearing rings. The test machine was an universal circular grinding machine type Studer S40.

ROUGH GRINDING OF BEARING RINGS

Prior to the superfinishing process a rough grinding process was performed to achieve the

desired profile of the race of outer ring. A vitrified bond Al_2O_3 grinding wheel (93A80 H13VP601) grain size 185 μm from Winterthur Technology Group was used for rough grinding. Dressing was performed with a dressing diamond. The dressing depth of cut a_{ed} was 5 μm the radial feed speed v_{fad} was 500 mm/min. The cutting speed was the same in dressing as in grinding $v_c = 50$ m/s. The workpiece rotation speed was $v_w = 0.55$ m/s leading to a speed ratio of about $q = 90$. Rough grinding was performed in three process steps with varying feed speed. Roughing was performed with a feed speed $v_{fr} = 0.08$ mm/min, finishing $v_{fr} = 0.03$ mm/min and fine finishing with $v_{fr} = 0.01$ mm/min. After rough grinding the bearing rings had an average roughness of $R_z = 1.84$ μm , $R_k = 1.09$ μm , $R_{pk} = 0.42$ μm and $R_{vk} = 0.78$ μm . The radial deviation was about 4.83 μm .

SUPERFINISHING

Different elastic bonded grinding wheel specifications from Artifex were proved to find the proper specification to investigate the influence of process parameters on surface quality (table 1). Each grinding experiment was repeated three times to statistically firm the results.

TABLE 1. specifications of elastic bonded grinding wheels

Name	Grain type	Grain size	
		FEPA	μm
A	White Aluminum oxide	400	17.3
B	White Aluminum oxide	600	9.3
C	Regular Aluminum oxide	800	6.5
D	Regular Aluminum oxide	800	6.5
E	Regular Aluminum oxide	800	6.5
F	White Aluminum oxide	800	6.5

The grinding wheels had a polyurethane bond with varying hardness. The hardness of the specifications A, B, E and F was equal. Specification C had a higher hardness and D had the highest hardness of the investigated

bonding types. The grinding and dressing parameters were kept constant while investigating the best grinding wheel specification. Dressing was performed using a dressing diamond. The overlapping rate in dressing was $U_d = 9.6$, the depth of dressing cut was $a_{ed} = 2.5 \mu\text{m}$. The cutting speed $v_c = 36 \text{ m/s}$ was equal in dressing and cutting. The workpiece rotation speed v_w was 0.7 m/s , the feed speed $v_{fr} = 0.05 \text{ mm/min}$, the total infeed was 0.15 mm over the diameter and spark out time 5 s .

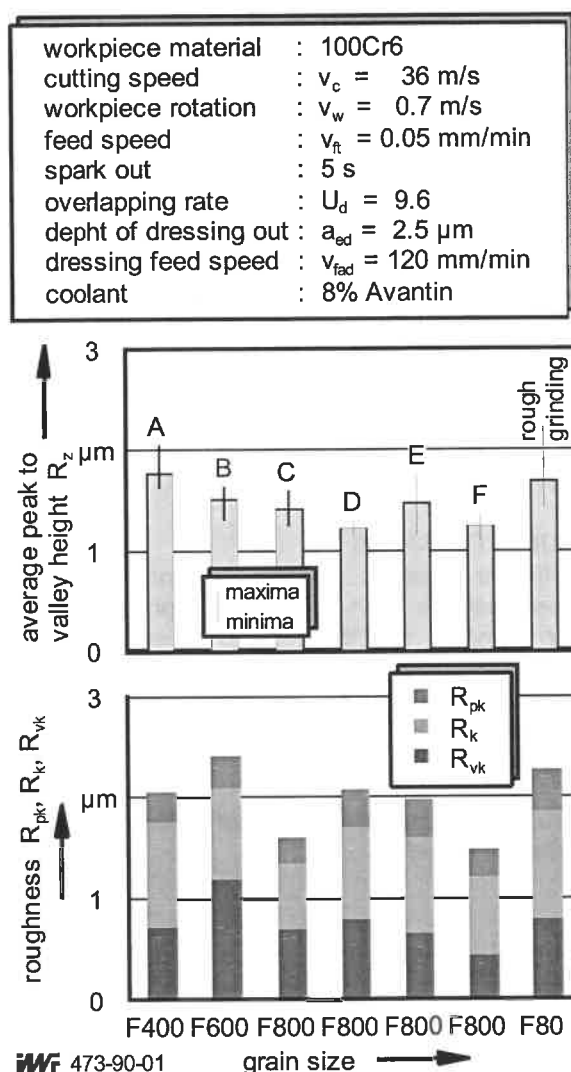


FIGURE 1. roughness values after superfinishing with elastic bonded grinding wheels

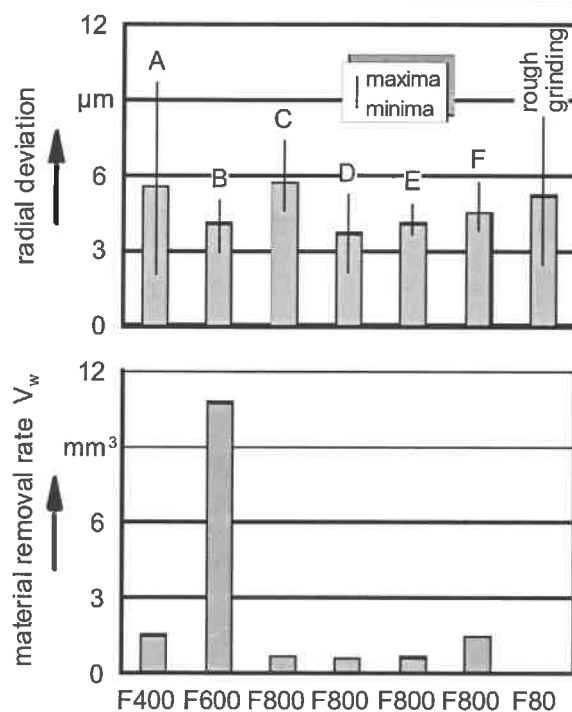
Usually, a grinding process is characterized by the infeed speed regardless the total infeed. However, using the elastic bonded grinding wheels, the total infeed becomes more

important. Due to the elasticity of the grinding wheel, the total infeed cannot be reached, though, with increasing infeed the force between grinding wheel and workpiece rises, which influences the surface quality. The goal is to reach an average peak to valley heights of under $R_z < 1 \mu\text{m}$, which is equal to the values reached in conventional superfinishing processes [3]. Although, this goal is not reached yet, the first results are promising. The smallest roughness values were achieved using the grinding wheel specification D and F (figure 1). It can be seen, that all kinds of grinding wheels achieved a reduction of core roughness depth R_k and reduced peak height R_{pk} . Thus, the wear resistance of the surface was improved compared to rough grinding. Regarding grain size, it could be seen, that the surface roughness was improved with smaller grain size. The type of grain did not influence the roughness noticeably. Investigating the differences in bonding hardness, the lowest average peak to valley height R_z was reached with the hardest bond type. Though, the smallest core roughness depth R_k and reduced peak height R_{pk} were reached using the bonding type with medium hardness. Regarding tribological behavior of the surface, the medium bond hardness has to be preferred since roughness peaks are removed more efficiently and the peak to valley height R_z is only slightly worse.

A further function of superfinishing is to realize the required roundness accuracy of bearing rings. The radial deviation was worsened using the grinding wheel with white aluminum oxide F400 (A) and using the regular aluminum oxide with medium bond hardness (C) (figure 2). Although, the best results were found using the grinding wheel with the hardest bond (D) the highest reduction in radial deviation was achieved using the grinding wheel with regular aluminum oxide F800 and comparatively flexible bond (E). The calculated material removal during superfinishing was 133.9 mm^3 . Though, the real amount of material removed during superfinishing was much smaller. A specific force between grinding wheel and workpiece needs to be applied to enable cutting of material. This needs to be considered during production, since the geometrical accuracy of the bearing ring is very important. The material removal reached with the aluminum oxide grinding wheel F600 (B) was noticeable higher than those of the other investigated elastic bonded grinding wheels. Nevertheless, it can be seen, that the

material removal is higher using white aluminum oxide compared to regular aluminum oxide, due to the higher sharpness of white aluminum oxide, while the bonding hardness did not influence the material removal rate.

workpiece material	: 100Cr6
cutting speed	: $v_c = 36$ m/s
workpiece rotation	: $v_w = 0.7$ m/s
feed speed	: $v_{ft} = 0.05$ mm/min
spark out	: 5 s
overlapping rate	: $U_d = 9.6$
depth of dressing out	: $a_{ed} = 2.5$ μ m
dressing feed speed	: $v_{fad} = 120$ mm/min
coolant	: 8% Avantin

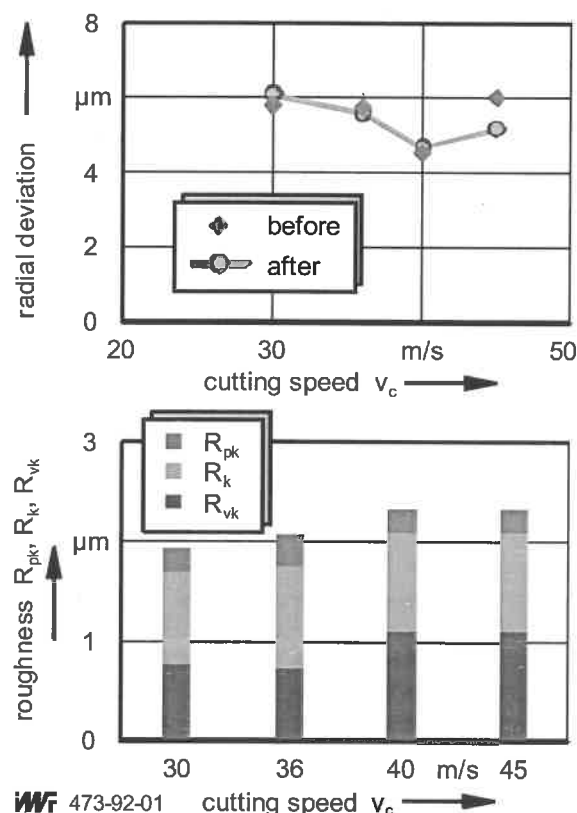


Wf 473-91-01 grain size $\rightarrow$
FIGURE 2. Influence of grinding wheel specification on radial deviation and material removal

To investigate the influence of cutting speed the grinding wheel specification A was used. The average peak to valley heights R_z , core roughness depth R_k and reduced peak height R_{pk} was hardly affected by cutting speed. The reduced valley height seems to rise with cutting speed, since the centrifugal force rises with rotation speed and thus, the elastic bond acts harder than with lower cutting speed. It can be seen that the radial deviation is hardly influenced by the superfinishing step.

Nevertheless, a falling tendency is noticed with rising cutting speed in superfinishing. The highest reduction in radial deviation was achieved at the highest cutting speed (figure 3). There was hardly any material removed in the superfinishing step. The depth of cut reached was only 1 % of the supposed depth of cut. Thus, influences of cutting speed on material removal rate could not be seen.

tool	: A
workpiece material	: 100Cr6
workpiece rotation	: $v_w = 0.7$ m/s
feed speed	: $v_{ft} = 0.05$ mm/min
spark out	: 5 s
overlapping rate	: $U_d = 9.6$
depth of dressing out	: $a_{ed} = 2.5$ μ m
coolant	: 8% Avantin



Wf 473-92-01 cutting speed $v_c \rightarrow$
FIGURE 3. Influence of cutting speed

The influence of feed speed on superfinishing results was investigated using the wheel type B. Since the feed speed influences the force between grinding wheel and workpiece it should have a noticeable influence. Nevertheless, despite expectation, the feed speed did not influence the radial deviation and material removal rate. Though, it can be seen that the

radial deviation is improved by the superfinishing process. The roughness is slightly rising with the feed speed and the roughness reached in rough grinding could be improved. Though, regarding the parameters of the Abbott-curve, the feed speed does not seem to influence the tribological behavior of the bearing race (figure 4).

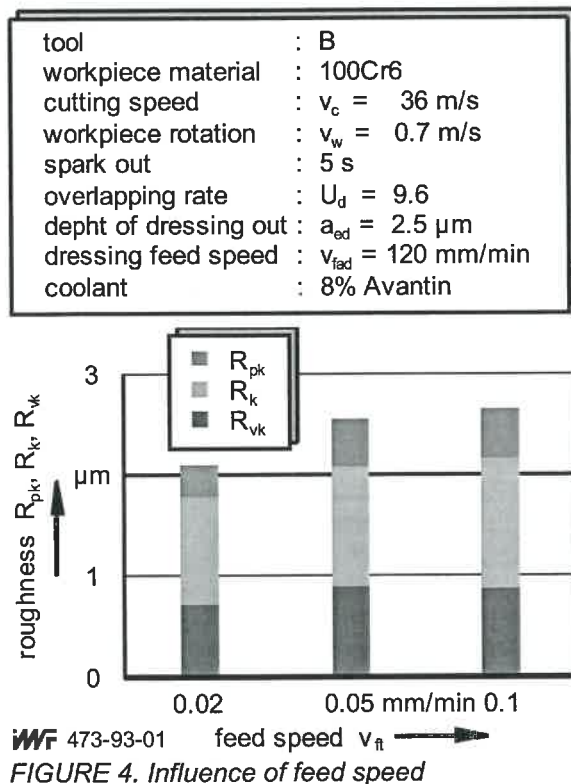


FIGURE 4. Influence of feed speed

So far, the experiments presented were conducted using an emulsion as coolant. It is known, that better surface qualities can be reached using grinding oil as coolant using conventional grinding wheels. To investigate if this influence can also be seen using elastic bonded wheels experiments with grinding wheel type D were conducted. As supposed, the roughness R_z was lower using oil ($R_z = 0.81$ μ m compared to 1.22 μ m) and the tribological behavior of the surface could be improved. The core roughness depth R_k and reduced peak height R_{pk} were about half as high using oil compared to emulsion while the reduced valley depth R_{vk} was slightly increased. Thus, oil should be used to conduct the superfinishing process. The material removal rate was five times higher using the cutting oil cause the friction between grinding wheel and workpiece is reduced leading to reduced bond wear.

CONCLUSION

The usage of elastic bonded grinding wheels for superfinishing of bearing rings was investigated. The best roughness values were reached using the smallest grain size F800. A higher bonding hardness leads to reduced radial deviation. An influence of cutting speed on bearing ring quality could not be found. The roughness R_z was improved with lower feed speed though, the feed speed did not influence the tribological behavior. The roughness was improved using oil as coolant compared to emulsion. Furthermore, the material removal rate was enhanced.

ACKNOWLEDGEMENT

We thank the Artifex Dr. Lohmann GmbH & Co KG for supporting the investigations on superfinishing of bearing rings.

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Subcomponent Developments for a Vortex Machining Test Facility

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INTRODUCTION

The goal of this research is to deterministically control sub-millimeter sized vortices produced by oscillating probes submerged in abrasive slurries to remove material in highly localized regions. Probes of varied diameters (7 μm to 0.5 mm), oscillating at high frequencies (over 100 Hz and up to 40 kHz) will produce localized vortices of varying size and magnitude. Slurry vortices generated at a specimen surface have been shown to produce material removal (MR) footprints with sub millimeter lateral dimensions [1]. The size of the process footprint will be controlled by the amplitude and frequency of the driving oscillations, and the orientation of the probe with respect to the workpiece. In addition to giving a brief overview of the theory, recent developments and component evaluations in the following areas will be discussed:

- Energy density calculations of flows induced by various machining conditions.
- Integration of xy and z translational stages for reliable collocation of probe and workpiece in the static slurry testing facility.
- FPGA-based, digital lock-in and phase locking capabilities for real-time probe characterization and control.
- Micro flow management system for controlled flows of larger, non-colloidal polishing slurries.
- Initial results and evaluation of static slurry testing.

BACKGROUND

FIGURE 1 depicts the basic concept of vortex machining. A slender oscillating probe is submerged into an abrasive slurry bath and positioned close to the workpiece surface. The probe oscillations induce localized vortices close to the workpiece surface. These vortices accelerate the slurry, and the abrasives within, across the workpiece surface removing material in the vortex vicinity. Undoubtedly both chemical and mechanical aspects contribute towards material removal; however initial focus is on the impact of the vortex dynamics.

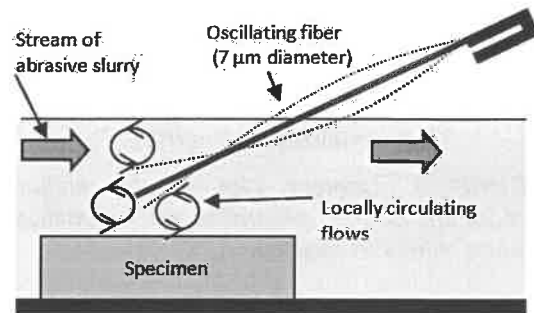


FIGURE 1. Schematic depicting the dynamics of vortex machining process [1].

THEORY

To model the vortex dynamics, a MatLAB code was written based on a solution to the Navier-Stokes equation for oscillating flows around a cylindrical solid (the probe) produced by Holtsmark et al. [2]. This solution can be used to generate the zero order flow field as well as the steady flow pattern that is established in the region around an oscillating probe centrally placed in a cylindrical reservoir. Modeling the stream functions will enable the prediction of the flow pattern and expected particle velocities (and, therefore, energies). While a more complete explanation can be found in [2], it is of interest to mention that the resulting streamlines consist of counter-rotating inner and outer vortices, see FIGURE 2. It is theorized that the faster-moving inner vortices contribute more significantly to the material removal while the outer vortices are responsible for bulk slurry flow. In general, the flow pattern is characterized by the dimensionless frequency number R_o , given by

$$R_o = a \left(\frac{\omega}{\eta} \right)^{1/2}$$

where a represents the radius of the cylinder, ω the oscillation frequency and η is the kinematic viscosity ($1 \times 10^{-6} \text{ m}^2 \text{ s}^{-1}$ for water at 20 °C). Typically, the size of the inner vortices decreases rapidly with an increasing dimensionless frequency number.

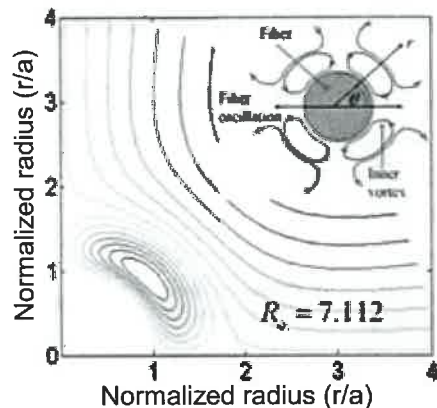


FIGURE 2. Contour plot of the stationary component of the streamline as a function of dimensionless frequency.

In addition, work has been done to predict the resulting flow energies associated with different probe diameters, amplitudes, and frequencies. The MatLAB code discussed previously was modified to plot energy density based on the solutions to the zero order and stationary flow fields. The energy density plot for a 7 μm diameter probe is shown in FIGURE 3. Current mathematical models are limited to cylindrical probes positioned orthogonally to the fluid reservoir. It is theorized that these energy calculations may correlate with experimental MR rates and footprint geometries.

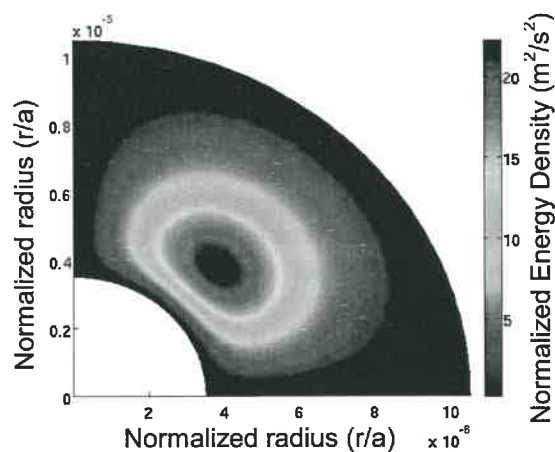


FIGURE 3. Energy density plot for a 7 μm diameter probe positioned orthogonally to fluid reservoir oscillating at 32.7 kHz with an amplitude of 25 μm ($R_o = 1.586$).

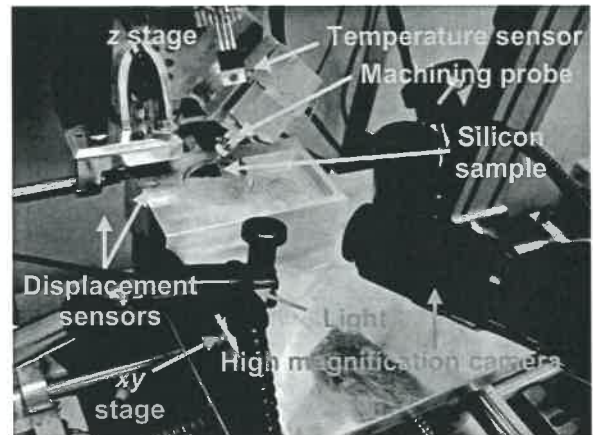


FIGURE 4. Picture vortex machining test apparatus with machining probe set at an angle of 45 degrees.

EXPERIMENTAL TEST APPARATUS

Previous experiments [1] provided promising results with material removal rates and surface roughness comparable to Elastic Emission Machining [3]. However, mechanical instabilities in the previous test facility resulted in multiple MR footprints. To improve the stability and control of this process, it was necessary to design and build a new test apparatus that can provide reliable collocation of the sample and oscillating probe. This has been achieved using a xy positioning stage for mounting the specimen and a z translation stage for the oscillating probe, see FIGURE 4. In addition, the tool and sample are secured using magnetically retained kinematic mounts.

The xy stage consists of two linearly actuated platforms powered with DC motors. The z stage consists of a long-range (150 mm) stepper actuator coupled to a short range (30 μm) piezoelectric actuated flexure. An additional rotary freedom is provided to set the angular orientation of the machining probe relative to the specimen. The stages are all controlled around a relatively compact metrology loop and have demonstrated stability of within $\pm 5 \mu\text{m}$ during testing, see FIGURE 5. In addition to position control, an FPGA-based, all-digital, lock-in amplifier with phase-locked loop has been implemented for real-time control and monitoring of the fiber actuators. This has facilitated long-term stability of probe oscillations and automated characterization using phase locking and frequency sweep capabilities.

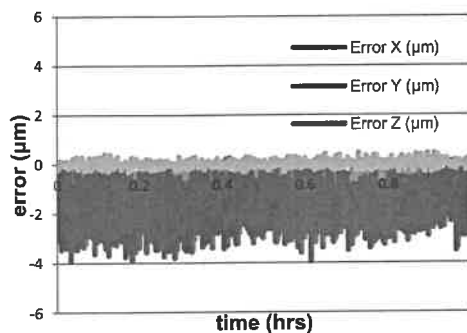


FIGURE 5: Graph of x , y , and z displacement error during a 1 hour machining test.

BROAD BANDWIDTH ACTUATOR

The probes in all tests to date have consisted of a $7\text{ }\mu\text{m}$ carbon fiber attached to a tuning fork oscillating at its resonant frequency of approximately 32.7 kHz [4]. The tuning fork oscillator provides adequate energy at resonance. However, it is not capable of providing substantial displacement at other frequencies. To explore a broader range of frequencies and amplitudes a separate actuation system, shown in FIGURE 6, consisting of a piezoelectric driven flexure actuator has been developed. This provides actuation across a 10 kHz bandwidth and is capable of driving probes with diameters from $7\text{ }\mu\text{m}$ to 0.5 mm . In addition, a knife-edge based, laser measurement system has been designed to provide real-time measurements of either the flexure actuator or directly from the probe shank. This will provide feedback for monitoring and control of machining conditions.

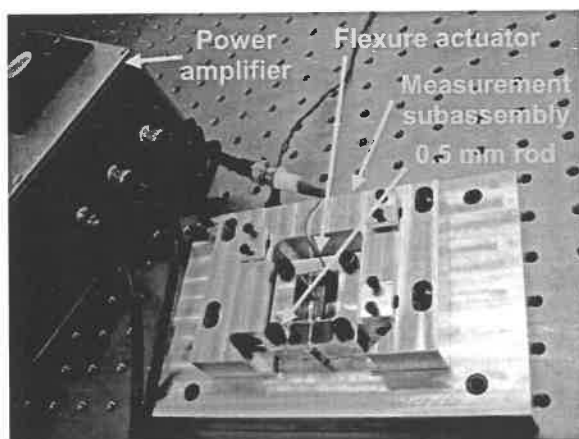


FIGURE 6. Picture of flexure actuator, laser measurement system, and high power piezoelectric amplifier.

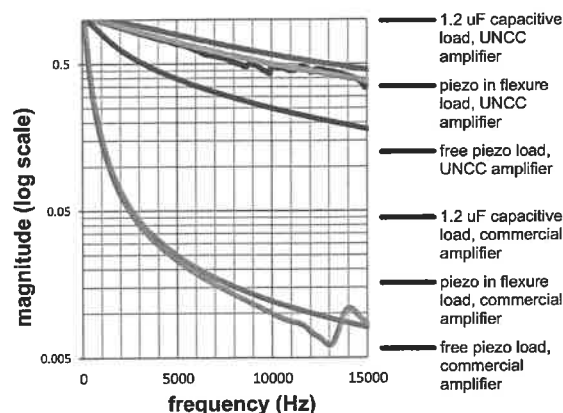


FIGURE 7. Graph showing the magnitude response of the UNCC power amplifier and commercial amplifier driving a $1.2\text{ }\mu\text{F}$ capacitor, piezoelectric actuated flexure, and unconstrained piezoelectric actuator. Piezoelectric actuator has an approximate capacitance of $1.4\text{ }\mu\text{F}$.

Because commercial piezoelectric amplifiers cannot deliver significant power to large capacitive loads over the bandwidth needed for our experiments, a high power amplifier has been designed and built. A comparison of the magnitude responses of the power amplifier and commercial amplifier driving various loads is shown in FIGURE 7. The piezoelectric actuator being used has an approximate maximum displacement of $15\text{ }\mu\text{m}$. However, significantly larger probe displacements are obtained when operating at resonant frequencies, see FIGURE 8.

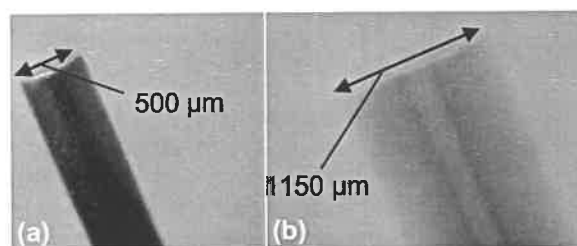


FIGURE 8. Pictures of (a) static $500\text{ }\mu\text{m}$ probe and (b) $500\text{ }\mu\text{m}$ probe at its first mode resonant frequency of 278 Hz with an amplitude of approximately $325\text{ }\mu\text{m}$.

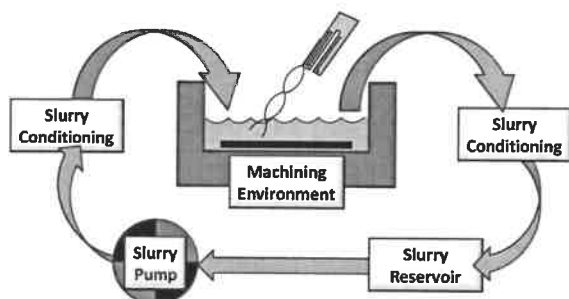


FIGURE 8. Block diagram including subcomponents of the slurry management system for the vortex machining test facility.

SLURRY MANAGEMENT SYSTEM

For stable testing over long time periods a laminar, low pulse, slurry management system is required. FIGURE 8 shows the subcomponent elements under development. The design process, which has been iterative in nature, is driven by results from both MR footprints and the in-situ measured gain and phase response from the oscillating fiber.

INITIAL TESTING RESULTS

Initial machining experiments have been pursued using the previously discussed test apparatus shown in FIGURE 4. In these tests, a $7\text{ }\mu\text{m}$ probe was positioned 45° with respect to the workpiece. The probe was held approximately $20\text{ }\mu\text{m}$ above the silicon surface in a static bath of 50% 0.05 μm Alumina slurry. The surface topography was measured using a Zygo, NewView 5000 white light interferometer.

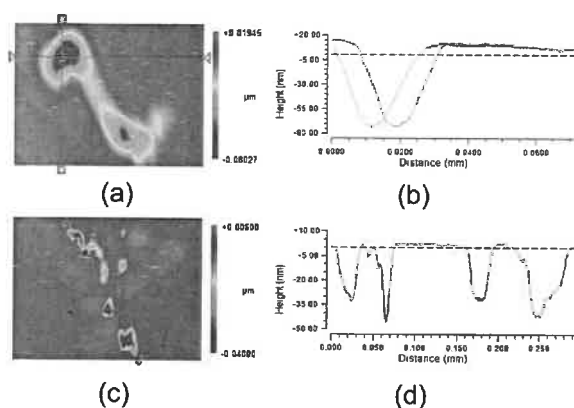


FIGURE 9. Surface topography of material removal footprint after 1 hour and 3 hour tests: (a) contour plot of 1 hour test ($70\text{ }\mu\text{m} \times 50\text{ }\mu\text{m}$ plot size), (b) profile of upper left footprint shown in (a), (c) contour plot of 3 hour test ($360\text{ }\mu\text{m} \times 270\text{ }\mu\text{m}$ plot size), (d) profile of multiple footprints shown in (c).

High evaporative rates combined with meniscus forces on the fiber caused the machining location to drift significantly during long term tests. As a result, repeatable footprints have been achieved through short term tests (< 1 hour), see FIGURE 9.

FUTURE PLANS

In the next few months, static slurry testing will continue and laminar flow tests with mixed, non-colloidal slurry will begin. Design has commenced on a high resolution, closed-loop slurry dispenser to compensate evaporation during static slurry testing. A separate, more rigid, machining frame is being constructed to facilitate investigations of larger diameter probes oscillating at frequencies varying from 100 to 10 kHz. These tests will aid correlation between flow patterns and energies with the resulting MR footprints.

ACKNOWLEDGEMENTS

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Effects of internal micro-defects on diamond tool wear in precision cutting - Micro FT-IR analysis of internal micro-defects at the tool edge-

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This paper details the relationship between tool wear properties and nitrogen impurities in diamond tools. Micro FT-IR analysis was carried out to determine the quantity of nitrogen impurities. The results show that nitrogen impurities significantly affect chipping and crater wear. Tools that contain more nitrogen impurities show less chipping. Crater wear volume corresponds to the quantity of nitrogen impurities in synthetic diamond tools and is affected by heat conduction and strength.

Key words: diamond tool, precision cutting, Micro FT-IR, nitrogen impurities, chipping, crater wear, redox reaction

1 INTRODUCTION

Diamond tools are essential devices for ultra-precision cutting because of their tool edge sharpness, low friction coefficient, and excellent hardness. Diamond tools produce an ultra-smooth surface finish but the life of the tools has proven to be extremely unpredictable. In copper ultra-precision cutting, Shimada et al have suggested that the tool life is almost completely determined not by the crater wear on the rake face, but by the surface finish. This is said to deteriorate at the worn tool edge and results from tool edge chipping [1]. They also report that the tool life and strength of diamond tools were can be controlled by nitrogen impurities [2].

Diamond is classified into four groups: Type Ia, Ib, IIa and IIb. Most natural diamonds belong to Type Ia. These have been used extensively for precision diamond cutting tools because the synthetic process has not been developed adequately from a cost viewpoint. However, recently, the synthetic diamond production technique have been significantly improved, and thus synthetic diamonds (Type Ib) have become a popular diamond tool material.

Nitrogen impurities have been shown to reduce tool life during copper cutting [2]. Type Ib diamond is reported to have a nitrogen content of around 10~100 ppm [3]. On the other hand, Type IIa contains less than 1 ppm. Therefore, it appears that a Type Ib tool should have a shorter tool life than a Type IIa tool.

However, the relationship between the nitrogen impurities and the tool life has yet to be determined. Thus, it is very important to analyze the nitrogen impurities at the tool edge, which is directly engaged in the cutting. In this paper, Micro FT-IR, which is suitable for this analysis, is employed to analyze the nitrogen impurities at the tool edge. From this analysis, the relationship between the nitrogen impurities and the tool life of the synthetic diamond tools (Type Ib and Type IIa) for copper precision cutting is discussed.

2 EXPERIMENTAL

Face-turning was carried out on a precision CNC lathe to investigate the wear of diamond tools. The work material was vacuum-annealed oxygen-free copper (OFC) cylindrical bars (99.96% purity) with a 62 mm diameter. This is the same material as that used for the

Table 1 Cutting conditions.

Work material	Oxygen-free copper (annealed)
Dimensions of work mm	$\phi 62 \times \phi 24 \times 10$
Tool material	Synthetic single crystal diamond
Rake angle α°	0
Clearance angle ϵ°	7
Nose angle γ°	130
Cutting form	Surfacing
Depth of cut t_1 μm	20
Feed rate f $\mu\text{m/rev}$	1
Cutting speed v m/min	123~283
Spindle speed N rpm	1500
Atmosphere	Dry

accelerating cells in a linear collider [4].

Table 1 shows the cutting conditions in the experiments. The tools used were synthetic single crystal diamond tools (Type Ib and IIa). Each tool had a 130° nose angle, 0° rake angle, and 7° clearance angle for ultraprecision cutting.

The depth of cut was $20\text{ }\mu\text{m}$, and the feed was $1\text{ }\mu\text{m/rev}$. Under these cutting conditions, the tool wear was uniform across the cutting edge. At each specified cutting distance, the depth of crater wear of the tool was measured with a laser microscope. Then, the crater wear volume was calculated from the depth of crater wear. In this paper, the tool life is determined by the sudden increase in machined surface roughness R_a .

3 EXPERIMENTAL RESULTS

3.1 Cutting performance of Type Ib and Type IIa

3.1.1 Machined surface roughness and tool life

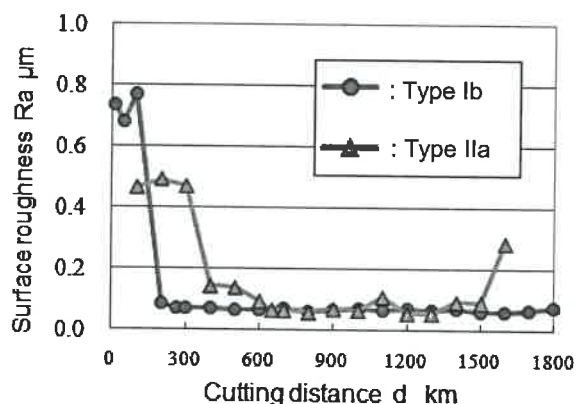


FIGURE 1 Relationship between surface roughness and cutting distance

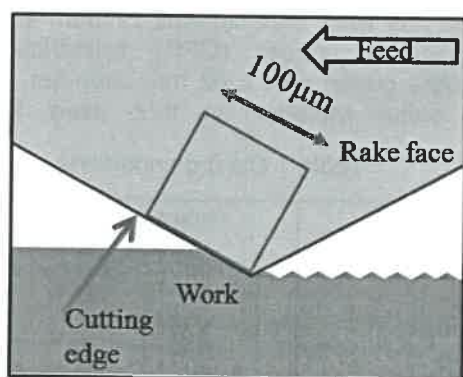


FIGURE 2 Cutting region and square analyzed by Micro FT-IR

Figure 1 shows the relationship between the surface roughness R_a and cutting distance d . A running-in cutting, which refers to the non-smooth cutting produced at the start of a cutting

job, was identified. The Type IIa tool showed a longer running-in cutting distance than that of the Type Ib tool. The Type IIa tool presented a sudden increase in the surface roughness at $d=1600\text{ km}$ which was recognized as the tool life. As for the Type Ib tool, a smooth surface was provided at $d=1,800\text{ km}$. Thus, the Type Ib tool showed a longer tool life than that of the Type IIa tool.

3.1.2 Absorbance of the diamond tool edge by Micro FT-IR

Figure 2 shows the cutting. The square area ($100 \times 100\text{ }\mu\text{m}$) along the cutting edge in the figure was analyzed by Micro FT-IR. Figure 3 shows the absorbance of the diamond tool edge analyzed by Micro FT-IR. The Type Ib tool had two peaks at 1130 and 1344 cm^{-1} which are referred to as the P1 center [5]. Existence of the P1 center indicates that an electrically neutral nitrogen atom is substituted for a carbon atom in the diamond lattice. Therefore, the Type Ib tool clearly contains nitrogen impurities. In contrast, the Type IIa tool has no peaks, and thus has many fewer nitrogen impurities.

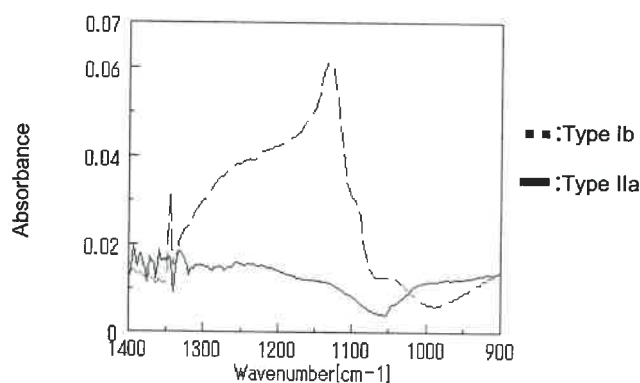
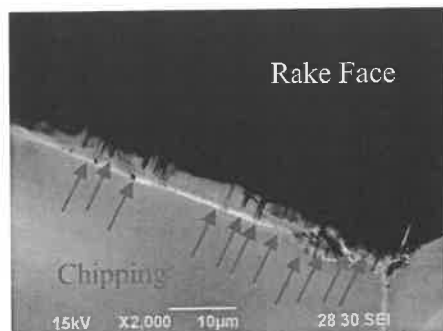


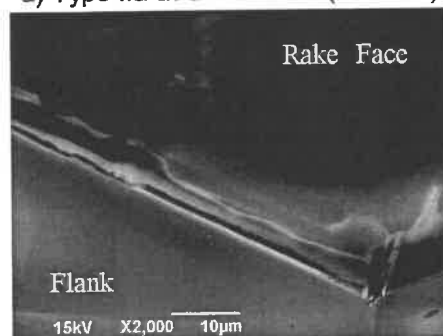
FIGURE 3 Absorbance at the diamond tool edge by Micro FT-IR analysis

3.1.3 Chipping on the tool edge

Figure 4 shows SEM images of the cutting tools. Figure 4 a) shows the Type IIa tool at the end of its tool life. A number of chippings can be seen at its edge. On the other hand, chippings are hardly seen on the Type Ib tool edge in Figure 4b. Chipping is a form of catastrophic break that occurs at the cutting edge. Taking into account the absorbance shown in Figure 3, it appears that an increase in nitrogen impurities results in a decrease in chipping, due to the suppression of crack propagation [2].



a) Type IIa at d=1600 km (Tool life)



b) Type Ib at d=1800 km

FIGURE 4 Chipping on the tool edge

3.2 Cutting performance of Type Ib tool

3.2.1 Machined surface roughness

In order to clarify the relationship between tool life and nitrogen impurities, we investigated the quantity of nitrogen impurities in three Type Ib tools using Micro FT-IR and compared the values from the viewpoint of tool life and wear.

Figure 5 shows the relationship between the machined surface roughness R_a and cutting distance d in Type Ib tools. The experiments were carried out up to the cutting distance $d=1,800$ km, but each tool produced a mirror surface finish. Therefore, it was impossible to compare tool life at this time.

3.2.2 Absorbance of the diamond tool edge by Micro FT-IR analysis

Figure 6 shows the absorbance of the Type Ib tool edge by the Micro FT-IR analysis. Peaks were found at the P1 center. The heights of the three peaks increased in the order: Tool 3, Tool 2, and Tool 1. The peak height is known to positively correspond to quantity of the nitrogen impurities, showing that the quantity of impurities in the tools increased in the same order.

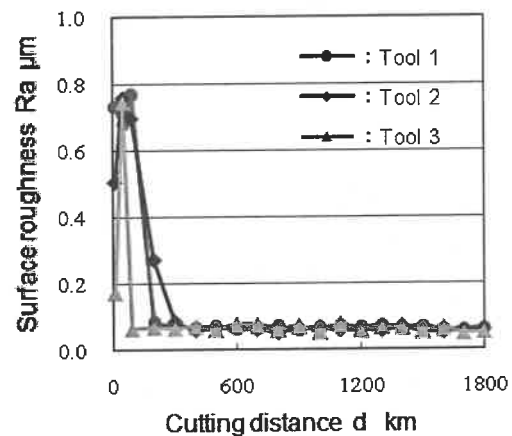


FIGURE 5 Relationship between machined surface roughness R_a and cutting distance

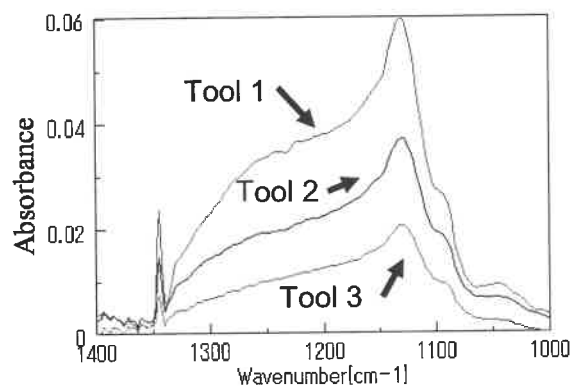


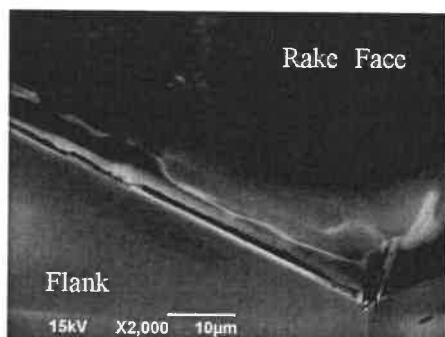
FIGURE 6 Absorbance at the diamond tool edge by Micro FT-IR analysis

3.2.3 Chipping on the tool edge

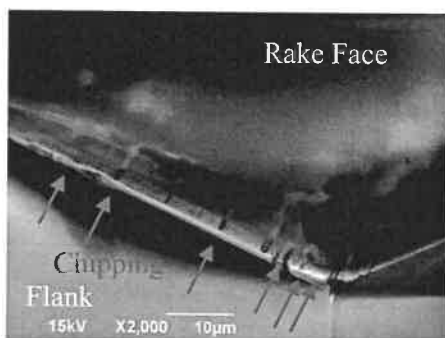
Figure 7 shows SEM images of the cutting tools of Type Ib. As shown in the figure, chippings were hardly seen on Tool 1. On the other hand, they were clearly visible on the Tool 3 cutting edge. Taking into account the absorbance in Figure 6, it seems that chipping decreased with an increase in nitrogen impurities.

3.3 Crater wear

Figure 8 shows changes in the crater wear volume with cutting distance. The Type IIa tool presented minimum crater wear volume and the crater wear volume increased in the order: Tool 3, Tool 2, and Tool 1. Taking into account



a) Tool 1



b) Tool 3

FIGURE 7 Wear pattern around cutting edge

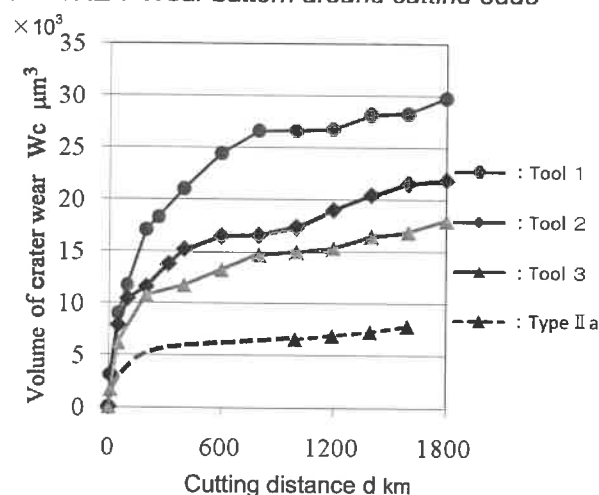


FIGURE 8 Changes in the crater wear volume with cutting distance

the FT-IR analysis shown in Figure 3 and Figure 6, the crater wear appears to correspond to the quantity of nitrogen impurities. The thermal conductivity in diamond is extremely high, but impurities are known to lower the conductivity. Phonon and conduction electrons serve as a media for thermal conduction in a solid body. Whereas conduction electrons play a major role in metals, phonons resulting from lattice vibrations play a major role in diamond.

Transmission of lattice vibrations, that dictate the amount of thermal conduction, can be retarded by nitrogen impurities contained in the diamond lattice, which results in a reduction in thermal conductivity.

In addition, nitrogen impurities can decrease the strength of diamond. Shimada et al. have reported that crater wear is a kind of the thermochemistry wear due to a redox reaction between the copper and carbon atoms that occurs during the copper cutting [6]. As the quantity of impurities increases, the cutting temperature will also increase due to the lower thermal conductivity, which then promotes the redox reaction.

4 CONCLUSION

- 1) Nitrogen impurities around the tool edge could be detected by Micro FT-IR analysis.
- 2) Micro FT-IR analysis indicated that Type IIa has fewer nitrogen impurities than Type Ib.
- 3) The Type Ib tool showed better wear properties than the Type IIa.
- 4) The Type Ib tool showed less chipping along the tool edge than the Type IIa tool.
- 5) Among Type Ib tools, those with more nitrogen impurities showed less chipping.
- 6) Crater wear volume corresponded to the quantity of nitrogen impurities in synthetic diamond tools, due to the heat conduction and strength.

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BAYESIAN UPDATING USING THE MARKOV CHAIN MONTE CARLO METHOD TO DETERMINE FORCE COEFFICIENTS IN END MILLING

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INTRODUCTION

Machining science, like many other technological fields, enjoyed tremendous gains in the 20th century. It was transformed from predominantly an empirical, trial and error discipline into one that now implements deterministic models for nearly every aspect of machining processes. In milling, for example, models are available to relate stability, part accuracy (from forced vibrations during stable machining), surface finish, and residual stresses to the selected operating parameters, material and tool properties, tool geometry, and part-tool-holder-spindle-machine dynamics. While the 21st century state-of-the-art continues to progress, obstacles remain. For example, all models include uncertainties in their inputs, as well as in the models themselves (due to the underlying assumptions and lack of knowledge). Also, the global marketplace has placed a premium on reduced production time and cost without sacrificing quality. Finally, implementation at the shop floor level can differ substantially from the research laboratory environment due to the availability of pre-process information.

To address these pressing issues, the authors are attempting to formulate milling, which is a critical value-added process in manufacturing, as a decision problem under uncertainty. As part of this work, efforts to implement Bayesian inference to determine the coefficients of a common force model used in end milling performance prediction are described.

MILLING FORCE MODEL

Milling forces can be described using the model provided in Eq. 1. In this equation, F_t is the tangential force component, K_t is the tangential cutting force coefficient, b is the axial depth of cut, h is the instantaneous chip thickness (which depends on the feed per tooth), K_{te} is the

tangential edge (plowing) coefficient, F_n is the normal force component, K_n is the normal cutting force coefficient, and K_{ne} is the normal edge coefficient [1].

$$\begin{aligned} F_t &= K_t b h + K_{te} b \\ F_n &= K_n b h + K_{ne} b \end{aligned} \quad (1)$$

The four coefficients in Eq. 1 are typically obtained by measuring the cutting forces with a dynamometer while machining at known axial depth and feed per tooth values. Measurements are performed over a range of feed per tooth values and a linear regression to the mean x (feed) and y direction forces is performed.

BAYESIAN INFERENCE

Bayesian inference models form a normative and rational method for updating beliefs when new information is available. Let the prior distribution about an uncertain event, A , at a state of information, $\&$, be $\{A|\&\}$, the likelihood of obtaining an experimental result B given that event A occurred be $\{B|A,\&\}$, and the probability of receiving experimental result B (without knowing A has occurred) be $\{B|\&\}$. Bayes' rule determines the posterior belief about event A after observing the experiment result B , $\{A|B,\&\}$ as shown in Eq. 2.

$$\{A|B,\&\} = \frac{\{A|\&\}\{B|A,\&\}}{\{B|\&\}} \quad (2)$$

The product of the prior and likelihood functions is used to determine the posterior distribution. In the case of multiple measurements, the posterior distribution after the first measurement or update becomes the prior for the second and so on.

BAYESIAN UPDATING OF MILLING FORCE COEFFICIENTS

In this paper, Bayesian updating using the Markov Chain Monte Carlo (MCMC) method to determine the force model coefficients in end milling is presented. As noted, Bayesian inference provides a systematic and formal way of updating beliefs when new information is available taking into account the uncertainty in variables. For the case of updating force coefficients using experimental force data, Bayes' rule is written as:

$$f_{K_t, K_n, K_{te}, K_{ne}}(K_t, K_n, K_{te}, K_{ne} | \bar{F}_{x,m}, \bar{F}_{y,m}) \propto f_{K_t, K_n, K_{te}, K_{ne}}(\bar{F}_{x,m}, \bar{F}_{y,m} | K_t, K_n, K_{te}, K_{ne}) \quad (3)$$

where $f_{K_t, K_n, K_{te}, K_{ne}}(K_t, K_n, K_{te}, K_{ne} | \bar{F}_{x,m}, \bar{F}_{y,m})$ is the posterior distribution of the force coefficients given measured values of the mean forces in the x and y directions, $\bar{F}_{x,m}$ and $\bar{F}_{y,m}$, $f_{K_t, K_n, K_{te}, K_{ne}}$ is the prior distributions of the force coefficients, and $l(\bar{F}_{x,m}, \bar{F}_{y,m} | K_t, K_n, K_{te}, K_{ne})$ is the likelihood of obtaining the measured mean force values given specified values of the force coefficients. In this notation, the subscript m denotes measured values from cutting experiments. The measured values were assumed to be statistically independent.

In the case of updating force coefficients as described by Eq. 3, the prior is a joint pdf of the force coefficients, K_t , K_n , K_{te} , and K_{ne} . As a result, the posterior is also a joint pdf of the force coefficients. In Bayesian inference, the MCMC technique can be used to sample from multivariate posterior distributions. The single-component Metropolis Hastings (MH) algorithm facilitates sampling from multivariate distributions without sensitivity to the number of variables [2-7]. To sample from a joint pdf, the algorithm samples one variable at a time and then proceeds sequentially to sample the remaining variables. The joint posterior pdf is the target pdf for MCMC. As noted, the posterior (target) pdf is the product of the prior and likelihood density functions.

EXPERIMENTAL RESULTS

This section describes the experimental setup used to perform force coefficient measurements. Experiments were performed using a 19 mm diameter inserted endmill (one square uncoated Kennametal 107888126 C9 JC carbide insert; zero rake and helix angles, 15 deg relief angle,

9.53 mm square x 3.18 mm). The workpiece material was 1018 steel. The cutting force was measured using a table mounted dynamometer (Kistler 9257B). The first test was completed at a spindle speed, Ω , of 2500 rpm with a 3 mm axial depth of cut and 4.7 mm radial depth of cut (25% radial immersion, RI). The force coefficients were evaluated by performing a linear regression to the mean x (feed) and y direction forces obtained over a range of feed per tooth values: $f_t = \{0.03, 0.04, 0.05, 0.06, \text{ and } 0.07\}$ mm/tooth. Figures 1 and 2 show the linear least squares best fit to the experimental mean forces in the x and y directions, respectively. The mean forces show a linear increase for both the x and y directions and the quality of fit is good ($R^2 = 0.99$). The force coefficients were determined using the slopes and intercepts from the data regression. The values of the experimental mean forces values are provided in Table 1. The force coefficient values calculated using the linear regression were: $K_t = 2149$ N/mm², $K_n = 1290.1$ N/mm², $K_{te} = 37.1$ N/mm and $K_{ne} = 37.1$ N/mm.

TABLE 1. Experimental mean forces in x and y directions for 25% radial immersion.

f_t (mm/tooth)	Mean F_x (N)	Mean F_y (N)
0.03	-11.50	40.13
0.04	-13.31	46.10
0.05	-14.83	50.03
0.06	-17.64	56.63
0.07	-19.10	62.06

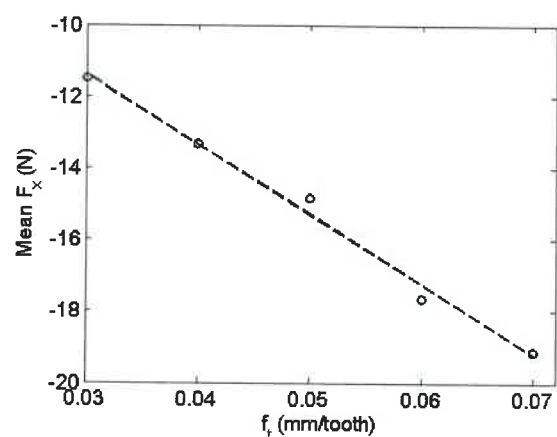


FIGURE 1: Linear regression to the mean forces in the x direction at 25% radial immersion.

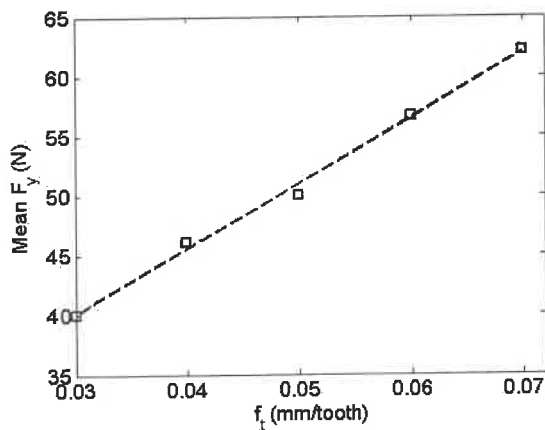


FIGURE 2: Linear regression to the mean forces in the y direction at 25% radial immersion.

The experimental force data listed in Table 1 was used to perform Bayesian updating on the force coefficients using the single component MH MCMC algorithm. The sampling was completed one coefficient at a time in the order $K_t \rightarrow K_n \rightarrow K_{te} \rightarrow K_{ne}$. The single-component MH algorithm facilitates sampling from the joint posterior pdf of the force coefficients, K_t , K_n , K_{te} , and K_{ne} . The posterior joint pdf was the target pdf for the MH algorithm. Table 2 lists the force coefficients obtained using the linear regression and Bayesian updating, where $N(\mu, \sigma)$ identifies the mean, μ , and standard deviation, σ , for the normal distributions.

TABLE 2. Force coefficient values obtained from linear regression and Bayesian updating using mean force values at 25% radial immersion.

	K_t (N/mm ²)	K_n (N/mm ²)	K_{te} (N/mm)	K_{ne} (N/mm)
Least squares	2149.0	1290.1	34.7	37.1
Bayes updating	$N(2116, 137.3)$	$N(1284, 130.2)$	$N(35.5, 3.2)$	$N(37.4, 3.2)$

For this analysis, the prior distribution of force coefficients was assumed to be a joint uniform distribution. Force coefficients were assumed to be independent for the prior. The marginal prior pdfs of the force coefficients were specified as: K_t (N/mm²) = $U(0, 3000)$, K_n (N/mm²) = $U(0, 3000)$, K_{te} (N/mm) = $U(0, 100)$, and K_{ne} (N/mm) = $U(0, 100)$, where U represents a uniform distribution and the parenthetical terms indicated the lower and upper values of the range. An uncertainty of 1 N standard deviation was assumed in the mean force data, which was

based on the user's belief regarding experimental uncertainty in measured force values. The mean force values were calculated using the current state of the chain for the specified cut geometry [1]. The likelihood for the x and y directions was the value of each pdf for the experimental mean forces.

A second test was completed at 50% RI with all other parameters the same. Figures 3 and 4 show the linear least squares best fit to the experimental mean forces in the x and y directions. The mean force in x direction does not show a clear linear trend (because it is approximately zero for a 50% RI and near the noise limit) and, therefore, the quality of fit is not good ($R^2 = 0.70$). The least squares fit to the y direction mean forces is very good ($R^2 = 0.99$), however. The cutting force coefficients, K_t and K_n , and the edge coefficients, K_{te} and K_{ne} , are not decoupled at partial radial immersions but depend on the slopes and intercepts of the least squares fits in both the x and y directions. Therefore, a poor fit in the x direction mean forces affects the values of all coefficients. Table 3 shows the mean forces.

TABLE 3. Experimental mean forces in x and y directions for 50% radial immersion.

f_t (mm/tooth)	Mean F_x (N)	Mean F_y (N)
0.03	1.51	63.35
0.04	1.11	74.71
0.05	0.93	84.98
0.06	0.67	95.29
0.07	-0.54	105.51

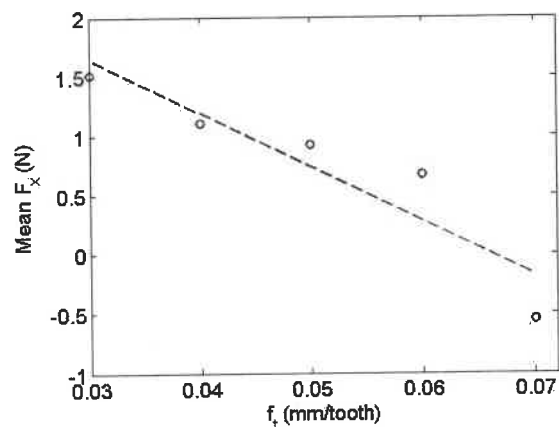


FIGURE 3: Linear regression to the mean forces in the x direction at 50% radial immersion.

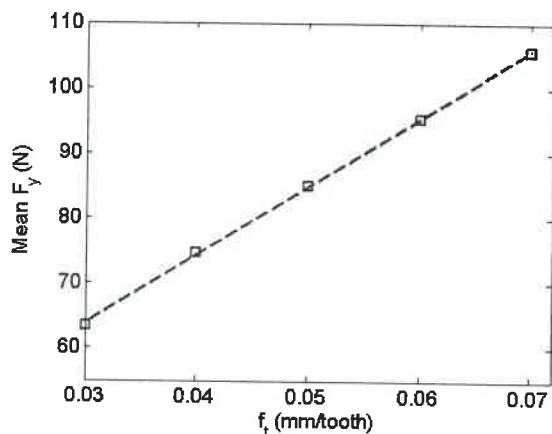


FIGURE 4: Linear regression to the mean forces in the y direction at 50% radial immersion.

The force coefficient values calculated using the linear regression were: $K_t = 2504.6 \text{ N/mm}^2$, $K_n = 1446.2 \text{ N/mm}^2$, $K_{te} = 37.5 \text{ N/mm}$ and $K_{ne} = 45.2 \text{ N/mm}$. The mean force data listed in Table 3 was also used to update the force coefficients distribution by the MCMC algorithm. Table 4 lists the force coefficients obtained using the linear regression and Bayesian updating. It is seen that the force coefficient values obtained from the two methods do not agree. This is due to the poor least square fit for the mean x direction force. Because the Bayesian updating does not rely on a curve fit, the posterior distributions are not affected by the quality of the fit and are insensitive to the radial immersion (as expected); for Bayesian updating, the posterior distributions obtained at 25% and 50% RI agree closely (see Tables 2 and 4).

TABLE 4. Force coefficient values obtained from linear regression and Bayesian updating using mean force values at 50% radial immersion.

	K_t (N/mm ²)	K_n (N/mm ²)	K_{te} (N/mm)	K_{ne} (N/mm)
Least squares	2504.6	1446.2	37.5	45.2
Bayes updating	$N(2052, 67.8)$	$N(1187, 68.9)$	$N(30.4, 2.3)$	$N(36.7, 2.6)$

CONCLUSIONS

Bayesian updating of the force coefficients using the Markov Chain Monte Carlo (MCMC) method was presented. The single component Metropolis Hastings algorithm of MCMC was used. Bayesian inference provides a formal way of belief updating when new experimental data is available. Bayesian updating gives a posterior

distribution that incorporates the uncertainty in variables as compared to traditional methods like the linear regression which give a deterministic value. By combining prior knowledge and experimental results, Bayesian inference reduces the number of experiments required for uncertainty quantification. Using Bayesian updating, a single test can give a distribution for force coefficients. The posterior distribution samples provide the covariance of the joint distribution as well. Experimental milling results showed that the linear regression approach did not give consistent results at 50% RI due to a poor quality of fit in the x direction mean forces, whereas Bayesian updating yielded consistent results at both radial immersions tested. Also, since Bayesian updating does not rely on a least squares fit, mean force data at different feed per tooth values is not required.

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PRODUCTION OF TiB_2 BASED Ti-B-C COATED TOOLS FOR CUTTING TITANIUM ALLOY

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INTRODUCTION

Ti alloys are attractive materials due to their unique high strength-weight ratio that is maintained at elevated temperatures and to their exceptional corrosion resistance. Therefore, they have proven to be important engineering alloys for advanced applications in aero-engine, airframe manufacture and biomedical industries [1,2]. However, the Ti alloys are classified as difficult-to-machine materials because of the high chemical reactivity and the low thermal conductivity. These characteristics of the Ti alloys lead to excessive chipping, premature tool failure and poor surface finish on the workpieces.

We attempted to improve the performance of cutting tools against the Ti alloy by coating of Ti-B-C thin films with high mechanical strength [3,4]. In this research, we carried out the experiments to optimize the deposition condition of the Ti-B-C thin films and also to improve cutting properties against the Ti alloy by the Ti-B-C coated tools.

EXPERIMENTAL

Figure 1 shows the schematic diagram of DC magnetron sputtering apparatus used for the Ti-B-C film preparations in Ar plasma. The Ti-B-C thin films have been prepared on silicon substrates and cemented tungsten carbide cutting tools by DC magnetron sputtering using a sintered TiB_2 or $\text{TiB}_2+10\text{wt.}\%$ B target and a C target. The input target power for the TiB_2 or $\text{TiB}_2+10\text{wt.}\%$ B target was fixed at 250 W. To control the amount of C in the films and to investigate the influence of C contents on the crystallinity, the hardness, the adhesive strength and the frictional properties of the films, the input powers for the C target were varied from 0 to 200 W at a constant substrate temperature of 350 °C. The residual pressure (base pressure) prior to deposition in the chamber was changed from 4×10^{-4} to 2×10^{-3} Pa. The film depositions were carried out at an Ar gas flow rate in the range of 50 to 90 sccm and under a working pressure around 2 Pa. The deposition time was varied to obtain the films with constant thickness. The film thicknesses were measured to be

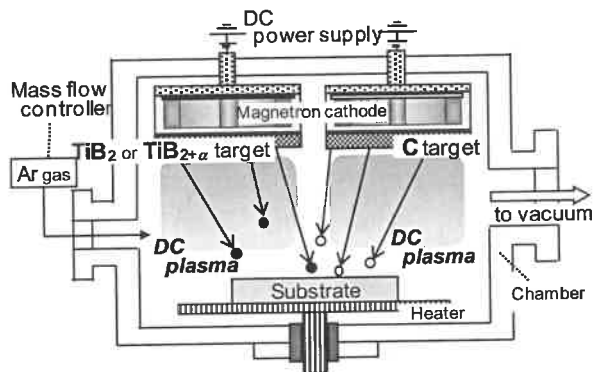


FIGURE 1. Schematic diagram of DC magnetron sputtering apparatus used for the Ti-B-C film preparations.

around 1 μm using a surface profilometer (DEKTAK IIA).

The composition of the films was investigated by X-ray photoelectron spectroscopy (XPS). In the XPS analyses, the binding energies of Ti 2p, B 1s and C 1s were measured from the surface of the films after Ar^+ ion sputter cleaning. X-ray diffraction (XRD) spectra were acquired at the X-ray incident angle 3° for evaluating the crystallinity of the Ti-B-C films.

Scratch tests for evaluating the adhesive strength of the films to the silicon substrates were made on a RESCA scratch tester CSR-02 device. Nano-indentation hardness measurements were carried out with a Berkovich diamond indenter using a commercially available equipment (Nano Indentation Tester ENT-1100a: ELIONIX). Maximum indentation load in the hardness measurements was 5 mN. The frictional properties of the Ti-B-C films were examined using a ball on disk friction tester in an ambient atmosphere. An AISI52100 ball of 5 mm in diameter was slid against the films at a normal load of 2.0 N and a sliding speed of 30 mm/s.

TABLE 1. Chemical composition of the TiB_2 and $TiB_{2+\alpha}$ films.

Target	Base pressure (Pa)	Ar gas flow rate (sccm)	Ti (%)	B (%)	O (%)	B/Ti
TiB_2	2×10^{-3}	50	34	56	8	1.6
		70	33	56	10	1.7
		90	32	55	12	1.7
$TiB_2 + 10\text{wt. \% B}$	2×10^{-3}	50	27	67	5	2.5
			25	67	7	2.7
			27	69	3	2.6
	2×10^{-3}	70	28	63	8	2.2
			27	66	5	2.2
			27	66	5	2.2

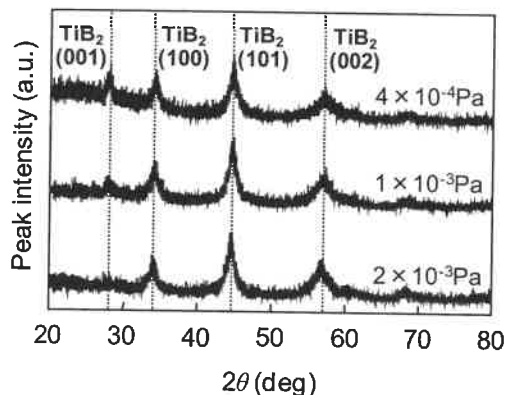


FIGURE 2. XRD spectra of the Ti-B-C films prepared with $TiB_2 + 10\text{wt. \% B}$ target at various base pressures.

Machining trials were carried out to evaluate the cutting performance of the Ti-B-C coated and the non-coated tools. The workpiece material used in the machining trials was a commercially available Ti-6Al-4V alloy. The dry machining experimental conditions were as following, cutting speed : 100 m/min, a feed rate : 0.1 mm/rev, depth of cut : 0.4 mm, cutting length of : 1000 m.

COMPOSITION AND CRYSTALLINITY OF THE FILMS

The chemical composition and the B/Ti ratios of the films prepared with the TiB_2 or the $TiB_2 + 10\text{wt. \% B}$ target are shown in Table 1. The TiB_2 films, prepared with the TiB_2 target, had the B/Ti ratios of below 2. On the other hand, the B/Ti ratios of the $TiB_{2+\alpha}$ films, prepared with $TiB_2 + 10\text{wt. \% B}$ target, were above 2, and the oxygen contents in the $TiB_{2+\alpha}$ films were about 5at.%, lower than those of the TiB_2 films. In the B rich films, oxidation of Ti was thought to be suppressed.

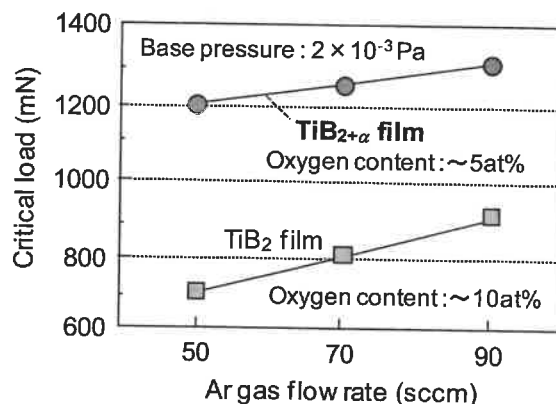


FIGURE 3. Critical loads of TiB_2 and $TiB_{2+\alpha}$ films prepared with various Ar gas flow rates.

Figure 2 shows the XRD spectra of the Ti-B-C films obtained with the $TiB_2 + 10\text{wt. \% B}$ target at various base pressures. The peaks of TiB_2 can be observed in all the XRD spectra of the films. It should be noted that the (001) peak came out for the film prepared at the base pressure of 4×10^{-4} Pa. Therefore, the crystallinity of the $TiB_{2+\alpha}$ films was improved with decreasing the base pressure.

ADHESIVE STRENGTH OF THE FILMS

Figure 3 shows critical loads in the scratch tests for the TiB_2 and the $TiB_{2+\alpha}$ films prepared at various Ar gas flow rates. The critical loads (the adhesive strength) of the $TiB_{2+\alpha}$ films were higher than those of the TiB_2 films irrespective of the Ar gas flow rates. In the $TiB_{2+\alpha}$ films, the oxygen contents were small. Therefore, the chemical activity of Ti in the film was maintained and it led to the high adhesive strength of the film.

MICRO HARDNESS OF THE FILMS

Micro hardness of the TiB_2 and the $TiB_{2+\alpha}$ films prepared at various Ar gas flow rates is shown in Figure 4. The $TiB_{2+\alpha}$ films had the higher adhesive strength, and the films with low oxygen contents had the higher hardness than the TiB_2 films. Figure 5 demonstrates the micro hardness of the $TiB_{2+\alpha}$ films prepared at various base pressures. The oxygen content in the $TiB_{2+\alpha}$ film prepared at the base pressure of 4×10^{-4} Pa was as low as 3at.%. The film had the highest micro hardness of 35 GPa. The improvement of the crystallinity with decreasing the base pressure is thought to be a reason for the high hardness of the film.

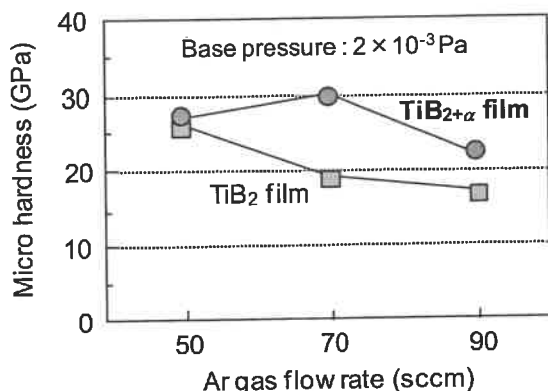


FIGURE 4. Micro hardness of TiB₂ and TiB_{2+α} films prepared at various Ar gas flow rates.

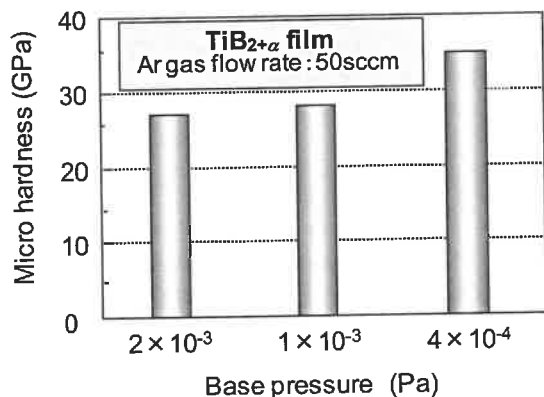


FIGURE 5. Micro hardness of TiB_{2+α} films prepared at various base pressures.

FRictional PROPERTIES OF THE FILMS

As described above, the TiB_{2+α} films had the high adhesive strength and the high hardness. However, its friction coefficient against a AISI52100 ball was about 0.7 as shown in Figure 6. Therefore, Ti-B-C film formation by C addition with the C target in the TiB_{2+α} film was tried to improve the frictional properties of the TiB_{2+α} films. The TiB_{2+α} based Ti-B-C films with C content around 50 % had the good frictional properties compared to that of TiB_{2+α} film and the friction coefficients were about 0.2 as shown in Figure 6. The Ti-B-C films with the low friction coefficient of 0.2 were slightly worn off. On the other hand, apparent wear was observed on the wear track of the TiB_{2+α} film with the high friction coefficient of 0.7. Therefore, the C addition is effective for improving the wear resistance of the TiB_{2+α} film as well as for reducing the friction coefficient.

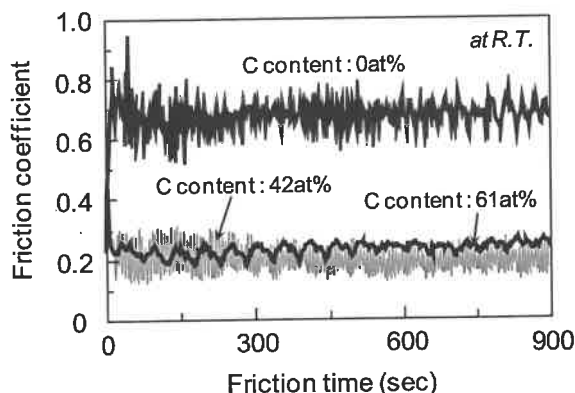


FIGURE 6. Friction coefficients of TiB_{2+α} based Ti-B-C films at room temperature.

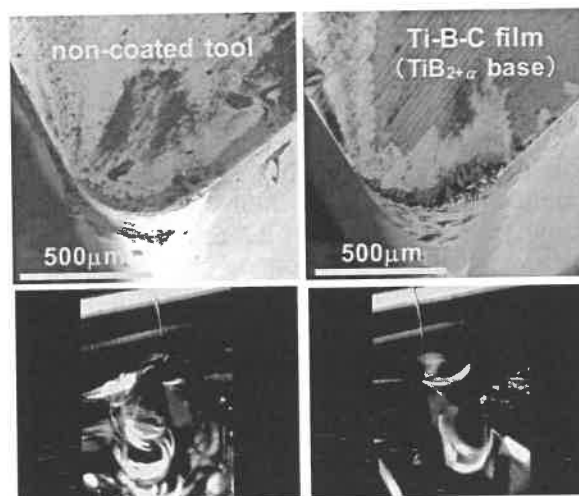


FIGURE 7. Tool wear and chip during machining for Ti alloy with Ti-B-C coated and non-coated tool, (depth of cut : 0.4 mm, cutting length : 1000 m).

CUTTING PROPERTIES OF THE Ti-B-C COATED TOOL

Cemented tungsten carbide cutting tools were coated by the Ti-B-C film with high adhesive strength, high hardness and low frictional properties. Figure 7 shows the tool wear and the chip formation when machining the Ti alloy by the non-coated and the TiB_{2+α} based Ti-B-C coated tool at the cutting speed of 100 m/min. The chip formation process became smooth by using the Ti-B-C coated tool. The chip formed with the Ti-B-C coated tool was smoothed as compared to that formed with the non-coated tool as shown in Figure 8. Furthermore, the cutting force when machining the Ti alloy

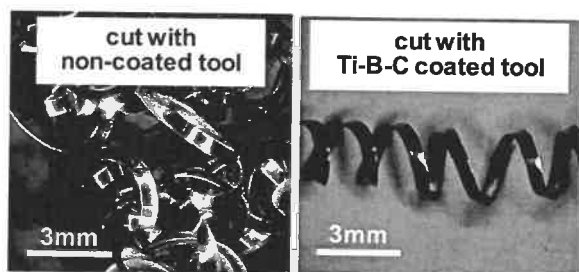


FIGURE 8. Chip in the machining of Ti alloy with Ti-B-C coated and non-coated tool, (depth of cut : 0.4 mm, cutting length : 1000 m).

became low by using the Ti-B-C coated tool, and the surface of the Ti alloy was smooth after cutting with the coated tool. However, the Ti-B-C coating on the edge of the cutting tool was peeled off at the cutting length of 1000 m as shown in Figure 7.

FUTURE WORK

In this study, the Ti-B-C films with the high adhesive strength, the high hardness and the low frictional properties can be obtained. However, the Ti-B-C coating on the cutting tool was peeled off at the cutting length of 1000 m. Therefore, the adhesive strength of the Ti-B-C films must be further improved for applying to

the cutting tool against the Ti alloy. Ti interlayer formation between the cutting tool and the Ti-B-C film may be effective for improving the adhesive strength of the films.

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OPTIMIZATION OF ELLIPTICAL VIBRATION ASSISTED MACHINING PROCESS THROUGH FINITE ELEMENT MODELING

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INTRODUCTION

Ultrasonic elliptical vibration assisted machining (EVAM) was introduced in the mid nineties as an improved method for ultraprecision machining [1]. EVAM applies micrometer scale elliptical motion to a tool tip that has shown to reduce tool forces and improve tool life and surface finish. It is also used to machine workpiece materials, such as steel, that otherwise cannot be diamond turned due to excessive chemical tool wear. In addition to forming smaller chips, the EVAM tool periodically comes out of contact with the workpiece, thereby allowing it to cool between cuts and reduce the tool temperature. This is perceived as the reason for reduced thermochemical wear observed by researchers [2].

The complicated relationship between speed, depth of cut, frequency, ellipse shape, and tool temperature pose a challenging heat transfer problem. Finite element analysis of the EVAM process allows the dynamic, transient tool temperatures to be calculated. Metal cutting finite element software called AdvantEdge, by ThirdWave Systems Inc., provides the first available FE models of the EVAM process. A chemical wear model based on the Arrhenius equation for chemical kinetics has been previously introduced and used in conjunction with AdvantEdge tool temperature results [3]. This paper introduces further finite element modeling of the EVAM process. A parametric study is conducted to predict tool life as a function EVAM parameters in an effort to optimize the process. This information will be used in future EVAM actuator design.

WEAR MODEL

The same wear model used in [3] is utilized here. Maximum tool temperatures that occur in the vicinity of the tool tip are extracted from simulation data. Tool temperature vs. time is supplied to the Arrhenius wear model to get volumetric wear rate (dV/dt) vs. time. This model assumes the following form:

$$\frac{dV}{dt} = A \exp\left(\frac{-E_a}{RT}\right) \quad (1)$$

where V is the worn volume (cross-sectional wear area $\times$ width of cut), R is the gas constant (8.3144 J/mol), and T is the peak tool temperature (K). A and E_a , the pre-exponential constant and activation energy, are empirical constants specific to the interacting materials (diamond and steel in this case). These values were previously determined to be $4.5 \times 10^7 \mu\text{m}^3/\text{s}$ and 42.5 kJ/mol, respectively [3]. Average tool temperature is determined by integrating temperature over the entire time period. Since wear can only occur while the tool is in contact with the workpiece, the average wear rate, $(dV/dt)_{\text{Avg.}}$, is determined by integrating dV/dt over the tool-workpiece contact time, then dividing by the total vibration period. This can then be converted to wear per machining distance, dV/ds , by dividing by the workpiece velocity, $v_{\text{wkp}}.$ Though the Arrhenius model defines wear per time, wear per machining distance is more relatable to the desired area of workpiece surface to be machined.

EVAM requires extra machining parameters to be defined beyond those for non-vibration turning. A major constraint that limits the workpiece velocity is the theoretical surface finish in the upfeed (cutting) direction. This is determined similarly to predicted surface finish due to tool radius and cross-feed in conventional single point diamond turning, but uses ellipse major radius and upfeed per vibration cycle. Upfeed surface finish is determined by the following:

$$PV_{up} \approx \frac{b}{8} \left(\frac{v_{\text{wkp}}}{fa} \right)^2 \quad \text{for } \frac{v}{a\omega} \ll 1 \quad (2)$$

where a is horizontal (parallel to workpiece surface) ellipse amplitude, b is vertical ellipse amplitude, f is EVAM frequency, and v_{wkp} is workpiece velocity. Ellipse amplitudes a and b in this paper refer to $\frac{1}{2}$ the total ellipse size. This calculated PV term is used as a parametric constraint in the following FE studies.

An elliptical tool vibration routine was supplied by ThirdWave Systems for the AdvantEdge FE program in support of this project. Tool and workpiece materials (single crystal diamond and AISI 1118 steel) previously used to determine the Arrhenius wear model coefficients are also used in the following parametric studies. Figure 1 shows an example temperature contour plot of and EVAM simulation (110x20 μ m ellipse, v_{wkp} = 12 m/min, depth of cut = 40 μ m). Peak tool temperature vs. time is extracted from simulation results and supplied to the wear model previously described. Calculated wear rates are compared while varying ellipse shape, vibration frequency, and a combination of ellipse shape and tool frequency.

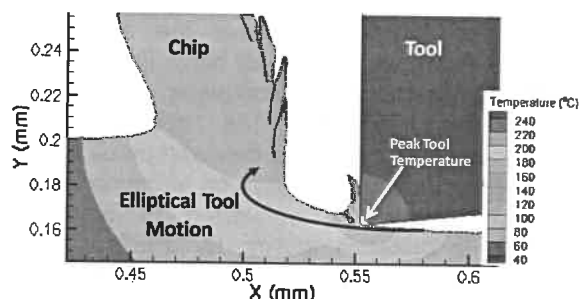


FIGURE 1. Example temperature contour plot of AdvantEdge EVAM simulations.

MODELS VARIING ELLIPSE SHAPE

Researchers have extolled vertically aligned ellipses ($b > a$) based on a phenomenon where friction on the tool reverses direction and 'lifts' the chip from the workpiece [1]. Others use horizontal ellipses ($a > b$) which allow higher machining speeds based on geometric surface finish. Three EVAM simulations were created in

AdvantEdge that apply horizontal, circular, and vertical ellipse shapes. Workpiece velocity was varied with the ellipse shape to maintain constant geometric PV surface finish. The horizontal and vertical simulations maintained the same ellipse size with differing orientation. The amplitude of the circular toolpath was chosen such that the swept area of the ellipse is the same for all three cases. Ellipse amplitudes and workpiece velocities are shown in Table 1. Workpiece material used was AISI 1118 steel, the tool was single crystal synthetic diamond, tool frequency was set at 20 kHz, depth of cut was 60 μ m, and ambient temperature was 20 C. The 60 μ m depth of cut ensured that ellipse center was below the workpiece surface for each simulation.

Simulations were run until tool temperatures reached semi-steady state. This was defined as temperature peaks differing less than 1% between consecutive vibration cycles. Resulting peak tool temperature profiles are shown in Figure 2 for single vibration cycles. Average wear rates were calculated using the aforementioned wear model and results are given in Table 1.

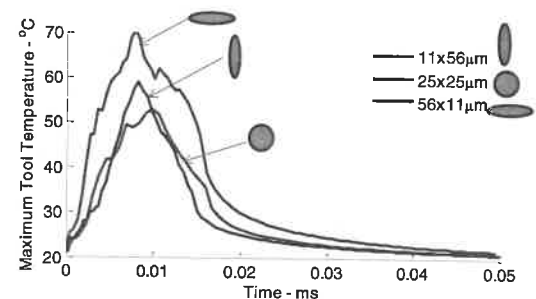


FIGURE 2. Temperature profiles during one EVAM cycle for varying ellipse shapes.

The horizontally aligned ellipse resulted in higher tool temperatures due to the higher relative velocity between the tool and workpiece. However, the available workpiece velocity with horizontal ellipse caused this wear to be 'spread

TABLE 1: Simulation parameters and wear results for FE simulations of EVAM varying ellipse shape. Frequency = 20 kHz, workpiece width = 1mm, and depth of cut = 60 μ m.

Ellipse Shape	Ellipse Amplitude, $a \times b$	Workpiece Velocity, v_{wkp}	Average Tool Temperature	Average Wear per Time, $(dV/dt)_{avg}$	Average Wear per Distance (dV/ds)
Vertical	11.3x56.6 μ m	14.22 m/min	28.3 C	1.30 μ m ³ /s	63.73 μ m ³ /m
Circular	25.3x25.3 μ m	4.20 m/min	29.0 C	1.35 μ m ³ /s	19.29 μ m ³ /m
Horizontal	56.6x11.3 μ m	1.22 m/min	34.3 C	2.46 μ m ³ /s	10.38 μ m ³ /m

TABLE 2. Simulation parameters and wear results for FE simulations of EVAM varying vibration frequency. Workpiece width = 1mm and depth of cut = 40 μm .

Frequency	Workpiece Velocity, v_{wkp}	Average Max. Tool Temperature	Average Wear per Time, $(dV/dt)_{avg.}$	Average Wear per Distance (dV/ds)
10 kHz	12 m/min	33.5 C	3.17 $\mu\text{m}^3/\text{s}$	15.85 $\mu\text{m}^3/\text{m}$
20 kHz	24 m/min	48.5 C	12.27 $\mu\text{m}^3/\text{s}$	30.68 $\mu\text{m}^3/\text{m}$
30 kHz	36 m/min	54.4 C	28.30 $\mu\text{m}^3/\text{s}$	47.17 $\mu\text{m}^3/\text{m}$
40 kHz	48 m/min	62.7 C	54.42 $\mu\text{m}^3/\text{s}$	65.53 $\mu\text{m}^3/\text{m}$

out' over further machining distances, effectively decreasing the wear per machining distance, dV/ds below that obtained from the vertical ellipse.

MODELS VARYING FREQUENCY

Higher EVAM tool frequencies allow higher workpiece speeds and reduced machine times. However, the relative velocity between the tool and workpiece, which partially determines the tool temperature, is increased with frequency. Four FEM cutting models were created that varied frequency with temperature results shown in Figure 3. Workpiece velocity was increased with frequency such that material removed per EVAM cycle was constant, as was the geometric PV surface. A horizontally aligned ellipse shape of 110x20 μm was chosen, depth of cut was 40 μm , and the same tool and workpiece material were used.

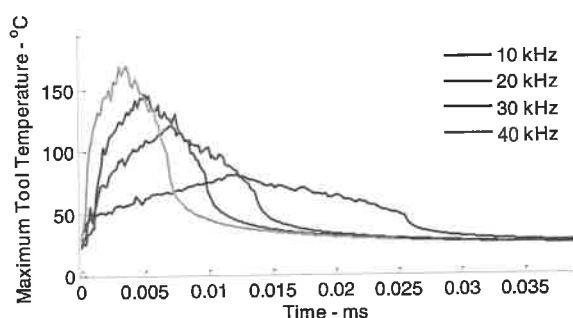


FIGURE 3. Temperature profiles during one EVAM cycle while varying tool frequencies.

As expected, peak and average tool temperatures increased with frequency due to the higher tool tip velocity. These values are given in Table 2. Wear per machining distance also increased with frequency. Unlike the simulations varying ellipse shape, the higher workpiece velocity was not enough to 'spread out' the wear at higher frequencies.

MODELS VARYING FREQUENCY AND AMPLITUDE

Ultrasonic EVAM actuators are most often designed as resonant devices. Utilizing higher resonant frequencies results in smaller vibration amplitudes at the same operating power. This same concept was applied to the EVAM simulations and horizontal amplitude was reduced at higher frequencies. Workpiece velocity was maintained constant for each simulation. The simultaneous increase in frequency and decrease in amplitude result in constant geometric PV surface for each simulation. Also, maximum horizontal velocity ($a\omega$) remains constant. While total material removal rate for each simulation is the same, material removal per EVAM cycle is less for the higher frequency and lower amplitude. Figure 4 shows temperature profiles from these simulations.

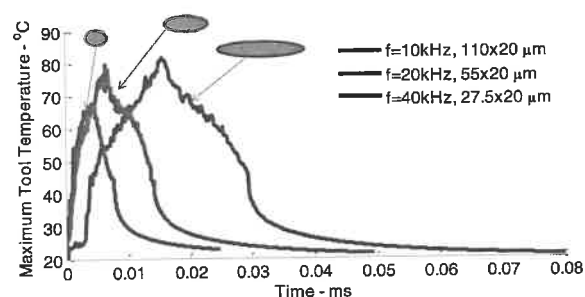


FIGURE 4. Temperature profiles during one EVAM cycle while varying tool frequencies and horizontal ellipse amplitude.

Maximum tool temperature averaged over time was highest for the 55x20 μm , 20 kHz simulation yet average wear rate was highest for the 110x20 μm , 10 kHz simulation. These values are presented in Table 3. Higher frequencies cause more temperature peaks over a certain time period. This increases the average temperature value determined by integrating the

TABLE 3. Simulation parameters and wear results for FE simulations of EVAM varying vibration frequency and ellipse amplitude. Workpiece velocity = 12 m/min, workpiece width = 1mm and depth of cut = 40 μm .

Ellipse Amplitude, axb	Frequency	Average Tool Temperature	Average Wear per Time, $(dV/dt)_{\text{avg}}$	Average Wear per Distance (dV/ds)
110x20 μm	10 kHz	33.5 C	3.17 $\mu\text{m}^3/\text{s}$	15.85 $\mu\text{m}^3/\text{m}$
55x20 μm	20 kHz	35.0 C	3.04 $\mu\text{m}^3/\text{s}$	15.19 $\mu\text{m}^3/\text{m}$
27.5x20 μm	40 kHz	34.0 C	2.35 $\mu\text{m}^3/\text{s}$	12.68 $\mu\text{m}^3/\text{m}$

peaks. Tool wear rate, dV/dt , increases nonlinearly with temperature and higher peak temperatures exacerbate the wear calculated by the Arrhenius model. The lower peak temperature of the 40 kHz simulation contributed to the minimal calculated wear rate of the three simulations.

Each set of machining parameters had the same workpiece velocity so wear being 'spread out' over further distance, such as the case for horizontal ellipses vs. vertical ellipses, does not play a role in this simulation set. Also, maximum horizontal tool tip velocities were the same for each simulation, though vertical components of velocity increased with higher frequency. The main reason for decreased peak temperature and wear rate for the 40 kHz simulation is the reduced volume removal per EVAM cycle.

DISCUSSION

Tool tip velocities and workpiece volume removal per EVAM cycle best determine tool temperatures. These factors are determined by the frequency, ellipse shape, workpiece velocity, and depth of cut. These studies assumed that geometric PV surface is a desired constraint, and EVAM parameters were varied accordingly.

Simulations that varied ellipse shape showed that ellipses aligned horizontal, or parallel to the workpiece surface, allow for longer tool life. This is possible since horizontal ellipses allow higher workpiece velocities to maintain the same PV surface finish. Temperature and wear per vibration cycle is higher for horizontal ellipses due to higher tip velocity and volume removed per cycle. However, the increased workpiece velocity allows wear levels to be spread out over further machining distance.

Simulations that varied frequency showed that tool tip velocities greatly affect temperature and

wear rate given the same chip shape or volume removed per cycle. However, the increased machining speeds allowed by the increased frequency are not enough to spread the wear over further machining distances.

Simulations that decreased ellipse amplitude while increasing vibration frequency allowed a constant horizontal tool tip velocity and workpiece velocity, but reduced the volume removed per cycle. This reduced the temperature and wear per vibration cycle;

Based on the thermo-chemical wear model and FE tool temperatures, tool life based on thermo-chemical wear per machining distance is improved when utilizing horizontally-aligned ellipses, higher tool frequencies, and reduced horizontal vibration amplitude. Though

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THROUGH-HOLE DRILLING OF SODA-LIME GLASS PLATE USING ELECTROPLATED DIAMOND TOOL

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INTRODUCTION

Hard and brittle materials such as ceramics and glasses have been widely used for functional materials [1-3]. The importance of machining technology for these materials is increasing, and the development of the machining technology is required. Since the machinability of these materials is very inferior to other traditional ones, the machining technology involves problems such as machining accuracy, machining productivity and machining cost. In particular, in through-hole mechanical drilling, cracks at the exit surface are the most important problem [4]. Recently, new applications in the electronics and aerospace industries require the drilling of small holes. Many researchers have studied various drilling methods. For example, the drilling of ceramics has been conducted using a core tool aided by ultrasonic vibration [5] and has been performed using an electroplated diamond tool [6]. However, the drilled hole is a diameter of more than 2mm and is a blind-hole.

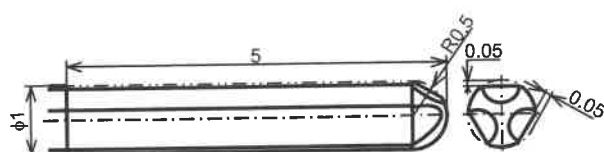
This paper deals with the through-hole drilling of

soda-lime glass plate using an electroplated diamond tool with a diameter of 1mm in order to find out drilling conditions for small crack.

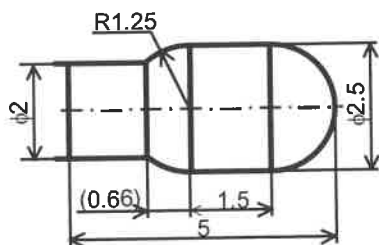
EXPERIMENTAL MEASUREMENT

Figure 1 illustrates the end form of electroplated diamond tool designed. The tools are composed of cylindrical body and hemispherical shape. The tool shank is carbon tool steel (JIS SK5). Diamond grains are electroplated from tool end to 5mm. The mesh size is #600 (average grain diameter is 28.5 μ m). Figure 1(a) shows the tool which has the end form of bowl with three pockets used in step drilling which called peck drilling or interrupted drilling. The tool attached planes were prepared in order to improve the chip removal. On the other hand, figure 1(b) shows the tool which has the end form of capsular form without the pockets. In the case of the tool with the pockets, the glass is broken and/or the tool is fractured.

Figure 2 shows the schematic drawing of experimental system. The machine tool used in the experiments is a machining-center (MD-45VA, Okuma Howa Co. Ltd.) with a Fanuc system. The machine tool uses an air turbine spindle (RSX, Big Daishowa Co. Ltd.) having an aerostatic bearing. Workpiece is soda-lime glass



(a) Bowl and three chip pockets



(b) Capsular form

Figure 1. Tool geometry

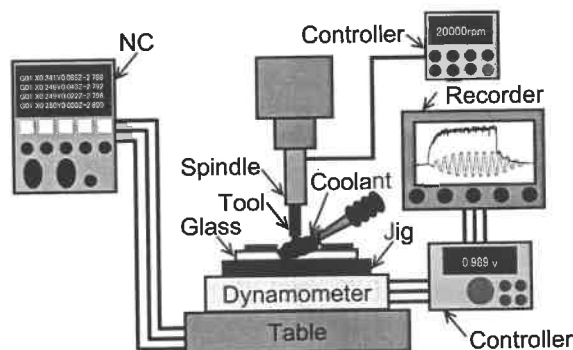


Figure 2. Schematic drawing of experimental system

plate which is being used as a plasma display panel. The glass plate is cut into rectangles by a glass cutter. The dimension of the test specimen is 100x50x2.8mm (length x width x thickness). The glass plate has a density of 2.74g/cm³, a coefficient of thermal expansion of 85x10⁻⁷/°C and a softening temperature of 829°C. Grinding fluid of emulsion type is diluted 10% with water. A dynamometer (9257B, Kistler Co. Ltd.) is mounted on the worktable. The thrust force measured by a dynamometer is transmitted to a digital recorder (NR-2000, Keyence Co. Ltd.). Tool failure pattern and crack appearance are observed using a scanning electron microscope (Hitachi High-Technologies Co. Ltd. S-4300) and a digital microscope (VHX-600, Keyence Co. Ltd.), respectively.

EXPERIMENTAL RESULTS

In step drilling, the drilling condition selected using quality engineering [7] was a spindle revolution of 15000min⁻¹, a feed speed of 1mm/min and a step feed of 0.1mm/step [8]. The step feed used is G83 code in a deep hole cycle on Fanuc system. In the deep hole cycle, the tool retracts all the way to the plunge clearance plane a number of times during the process to clear chips from the hole. Plunge clearance is the distance above an operation at which the tool starts to feed. The distance is 0.1mm. The tool will retract to this distance between step feed.

Figure 3 shows the variation of the maximum crack size in step drilling. The appearance of crack is observed in Fig.4 (a). The crack at the exit side is 0.1mm or less, but that at the entrance side is sometimes large. The crack size increases with the number of drilling, because the chip adheres gradually to the tool end. Chip adhered on the tool end is observed in Fig. 4 (b). Accordingly, the thrust force is also increasing with the number of drilling, as shown in Fig.5.

By the way, in Fig. 3 (a), compared with a drilling number of 50th, the crack size at a drilling number of 60th drilling is small. The thrust force decreases slightly at this time, as shown in Fig. 5. The reason is that the tool was cleaned by an ultrasonic wave after drilling of 50th. The crack size after drilling of 100th is 0.1mm and less. In addition, the thrust force increases gradually with drilling numbers. The observation of thrust force has clarified that the tool with chip pockets and bowl end is effective on the chip removal and crack restraint. Cleaned tool effects smaller crack size and longer tool life.

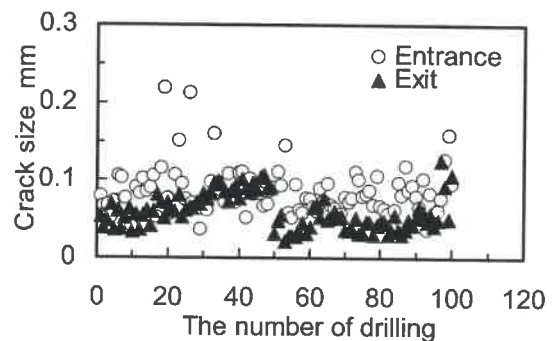


Figure 3. Variation of crack size in step drilling

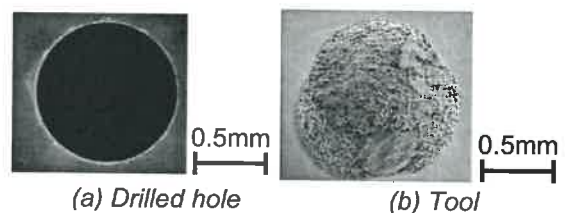


Figure 4. Appearance of crack at the exit hole and tool failure after drilling of 100th

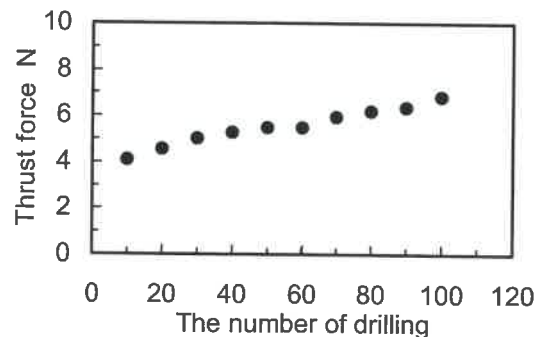


Figure 5. Progress of thrust force in step drilling

However, the drilling time takes about 430s per a hole due to a step feed of 0.1mm/step. Machining productivity is declining owing to the low step feed.

Figure 6 shows the progress of the crack size at the exit hole in the same type tool in attached planes of different sizes. The size of attached plane is 0.05mm (Tool A) or 0.1mm (Tool B). The crack size is large with increasing the number of drilling. The crack size of Tool B is slightly smaller than that of Tool A.

Figure 7 shows the maximum thrust force in Tool A and Tool B. The thrust force is increasing with increasing the number of drilling. The reason is that the diamond grains are covered by the chip. That is, the cutting edges blunt. The increase of tool B is slow. It is considered that the chip removal advantages by the attached

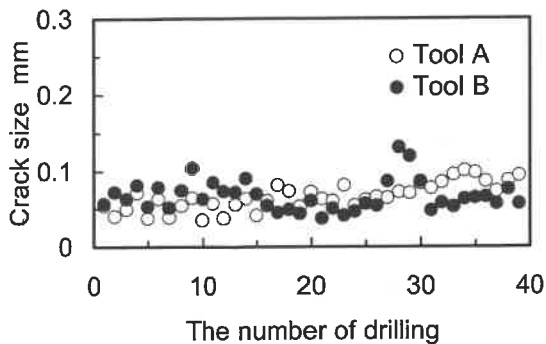


Figure 6. Comparison of the crack size at the exit hole between tool A and tool B

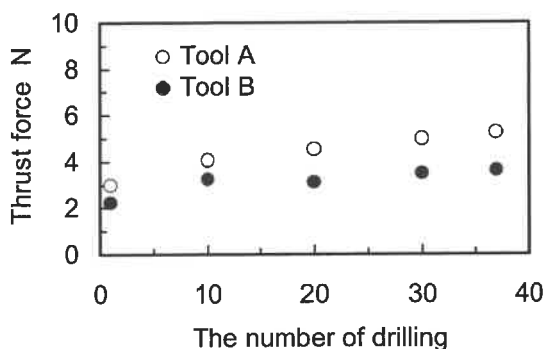


Figure 7. Comparison of the thrust force between tool A and tool B

large planes. The tool with planes is effective for chip removal, as imaged in Fig. 8. Machining productivity may be improved by refusing step feed.

In helical drilling condition using the capsular type tool is a spindle revolution of 20000min^{-1} , a feed rate of 20mm/min , an oscillating radius of 0.25mm and a helical pitch of 0.3mm [9]. The helical drilling is a superimposition of a circular interpolation and a linear movement. The screw-shaped movement will be executed with the constant helical pitch. The helical pitch will be programmed by the third parameter of the circular interpolation dependent on the selected plane. The space of chip removal is 0.25mm . Drilled hole diameter is 3mm . The drilling time takes about 90s per a hole. Compared with step drilling, machining productivity is progressive. However, the drilled holes have a large diameter. Figure 9 shows the relationship between the number of drilling and the crack size. The crack size at the entrance is very small. On the other hand, the crack size at the exit are $0.07\text{--}0.2\text{mm}$. Figure 10 shows the variation of the thrust force in helical drilling. Horizontal axis and vertical axis indicate the drilling time and the thrust force,

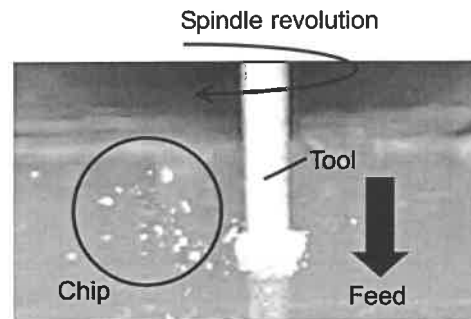


Figure 8. Observation of chip removal by high-speed video camera

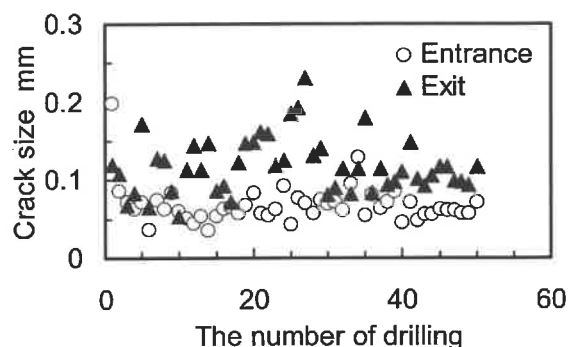


Figure 9. Progress of crack size in helical drilling

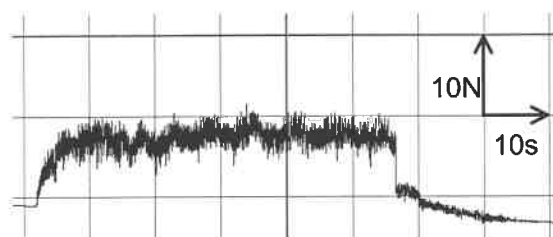


Figure 10. Variation of thrust force in helical drilling

respectively. The thrust force keeps the small rise in drilling. The large decrease of the thrust force shows that a hole has pierced. Large fracture occurs at this time. The maximum diameter of hole with large crack is smaller than that of tool. Small fracture occurs when the residual area is removed with the tool feed. Thereby, the crack around drilled holes is small. Figure 11 shows the relationship between the drilling depth and hole diameter. The hole has pierced, before the tool top reaches the backside of glass plate. The phenomenon is also observed in step drilling. Especially, in the case of helical drilling, the crack size at the pierced hole is large, as shown in Fig. 12. The thrust force increases with the number of drilling is increasing, as shown in Fig. 13. The

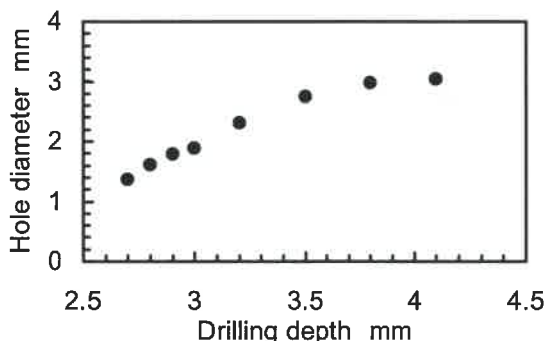


Figure 11. Relation between drilling depth and hole diameter

thrust force of helical drilling is larger than that of step drilling because of rapid feed speed. The faster speeds lead to the larger thrust force, and as a result the crack size becomes large.

CONCLUSION

This paper investigated tool geometries and drilling conditions to minimize crack size when drilling soda-lime glass plate with a thickness of 2.8mm using electroplated diamond tool. The obtained results are as follows.

The chip removal in the through-hole drilling is necessary in order to minimize crack size. In step drilling, the tool with chip pockets and bowl end is effective. In helical drilling, the tool with a space in drilling is effective. In each drilling method, the crack size increases with the drilling numbers. The thrust force rises owing to the adhered chip at the tool end. The small crack size is required to control the adhesive of chip. Therefore, the washing of tool is required in order to restrain crack. It can maintain the crack size of less than 0.1mm by removing chip adhered on the tool surface.

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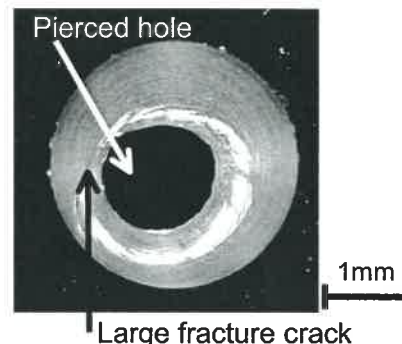


Figure 12. Appearance of pierced hole and large fracture crack from view of the exit surface at drilling depth of 2.7mm

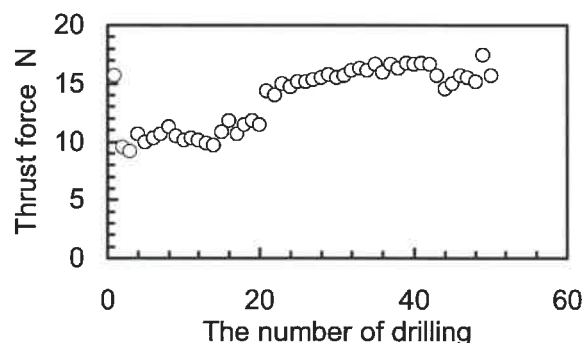


Figure 13. Progress of thrust force in helical drilling

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IMPROVEMENT OF FINISHING EFFICIENCY CONSIDERING CONTACT BEHAVIOR OF POLISHING PAD WITH WORKPIECE SURFACE

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INTRODUCTION

Recently the demand of a magnetic disk installed in hard disk drive (HDD) increases rapidly along with the popularization of the digital appliance. Then large capacity, downsized and low-cost magnetic disks are strongly required. In order to achieve the above-mentioned requirements, improvement of the polishing technologies of the magnetic disks is indispensable because the polishing process determines the quality and the production cost of the magnetic disks. Especially the achievement of further high finishing efficiency of glass disks as the substrates of magnetic disks is strongly required to decrease the production cost. Therefore, several polishing models concerning about the finishing efficiency have already been proposed [1, 2]. However, there are serious problems that the models can be applied only to specific polishing conditions.

In this study, we proposed a new polishing model considering the contact behavior between the polishing pad and workpiece surface based on the polishing experimental results and discussed key factors to increase the finishing efficiency.

INFLUENCE OF ABRASIVE CONCENTRATION ON MATERIAL REMOVAL RATE

The polishing experiments for glasses (ϕ 25.4 mm $\times$ t 1.0 mm) were carried out in a single-sided polishing machine (HYPREZ15, ENGIS Corp.), as shown in Fig. 1. Table 1 lists the polishing conditions. Three types of polishing pads, namely, nonwoven pad, suede pad and polyurethane foam pad were used.

When the polishing conditions were changed, concretely, the colloidal silica concentration of the slurry was changed, the conditioning was done with diamond dresser (#170) before

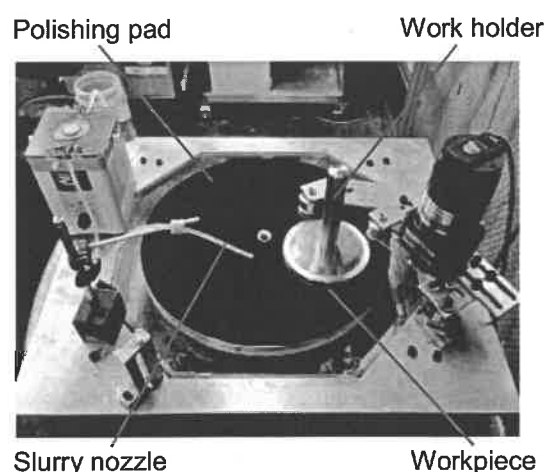


FIGURE 1. Apparatus of single-sided polishing machine.

TABLE 1. Polishing conditions.

Workpiece	Soda lime glass 4 pieces ϕ 25.4 mm - t 1.0 mm
Polishing pad	- Non woven pad - Suede pad - Polyurethane foam pad
- Diameter of pad	420 mm
Dresser	Diamond dresser (#170)
- Dressing pressure	9.8 kPa
Slurry	Colloidal silica
- Concentration	4, 8, 12, 16, 20, 24 wt%
- Supply rate	10 mL/min
Polishing pressure	30.7 kPa
Rotation of pad	40 rpm (26.4 m/min)
Rotation of workpiece	40 rpm (26.4 m/min)
Polishing time	20 min

polishing, and polishing experiments were carried out twice for 20 minutes. A finishing efficiency was calculated from the difference of

weight of workpiece before and after the polishing and the specific gravity.

Figure 2 shows the relationship between the material removal rate (MRR) and abrasive concentration in slurry. From this figure, it was found that MRR increased in proportion to the abrasive concentration in the usages of any polishing pads. The previously proposed polishing models [3, 4] cannot account for the result.

Generally, MRR is determined by the number of effective working abrasives and the load per abrasive. Then, the higher the abrasive concentration is, the higher the number of the effective abrasives is. On the other hand, the load per the abrasive decreases because the setting load acts on many abrasives, which leads to invariable MRR on the basis of the conventional polishing model [3, 4].

PROPOSAL OF POLISHING MODEL CONSIDERING CONTACT BEHAVIOR OF POLISHING PAD WITH WORKPIECE SURFACE

We have newly proposed polishing model considering polishing pad – abrasive – workpiece interfaces for overcoming the problem. In the conventional model, the setting load for workpiece acts on only abrasive – workpiece contact zone (Fig. 3). However the surface of polishing pad is considered to come in contact directly with workpiece surface due to the softness of the pad. Then the load acts on not only abrasive – workpiece contact zone but also polishing pad – workpiece one in the proposed model (Fig. 4).

In the proposed model, four types of contact zones between polishing pad and workpiece, namely, polishing pad – workpiece contact zone, abrasive – workpiece one (removal effect and not removal effect), and fluid – workpiece one exist (Fig. 4 (a) - (d)).

When the abrasive concentration becomes high, the abrasive flows into polishing pad – workpiece contact zone, and zone (a) replaces zone (b). The load per the abrasive is invariable because the load acting on zone (a) acts on zone (b) and the number of the effective abrasives increases. Thus, MRR increased in proportion to the abrasive concentration.

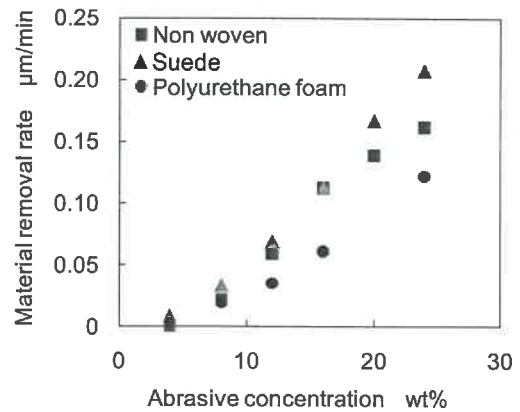


FIGURE 2. Material removal rate with increase of abrasive concentration.

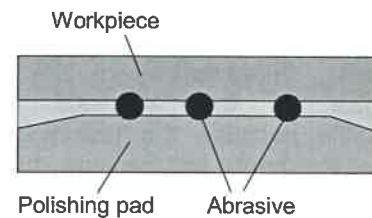


FIGURE 3. Conventional polishing model.

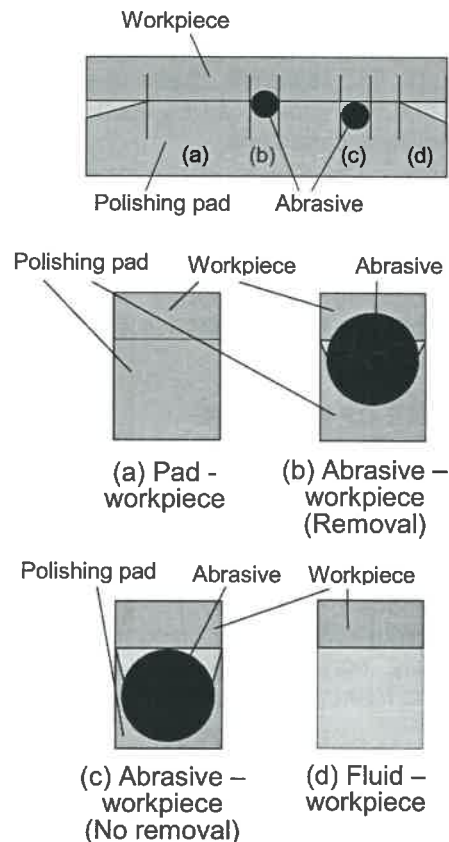


FIGURE 4. Proposed polishing model.

IMPROVEMENT OF MATERIAL REMOVAL RATE BASED ON THE POLISHING MODEL

Based on the proposed model, the increase of number of the effective abrasives, namely, the increase of zone (b) is effective for improving of MRR. Zone (b) can be increased by two methods. The basic one is to increase abrasive concentration in slurry. The other is to use a hard polishing pad.

When the abrasive flows into polishing pad – workpiece contact zone, zone (a) replaces zone (b) and zone (c). In zone (b), the abrasive supported to the polishing pad removes workpiece. On the other hand, in zone (c), the abrasive sinks completely in the polishing pad, and does not remove workpiece. It is decided whether zone (a) becomes zone (b) or zone (c) according to how much the abrasive sinks in the polishing pad (Fig. 5). Then, the hard polishing pad is thought to be suitable for achieving high MRR.

The polishing experiments with three kinds of suede pads having different hardness were conducted to examine the influence of the pad hardness on the finishing efficiency. Polishing conditions were basically the same as those listed in Table 1.

In order to measure the hardness of the polishing pad, the indenter made of the aluminum alloy (30 mm – 40 mm – t 20 mm) was put on the surface of the polishing pad. The amount of polishing pad deformation was measured with the dial gauge when load was applied within the range of 0 ~ 40 kPa. Figure 6 shows pressure and pad deformation. From this figure, it was found that the polishing pad was hard in the order of S1, S2, and S3.

Figure 7 shows the influence of hardness of polishing pad on MRR. From this figure, the finishing efficiency is high in the order of S1, S2, and S3, namely, when the polishing pad is hard, MRR becomes high.

Moreover, in S1, MRR became constant in high abrasive concentration (20, 24 wt%). The reason is thought to be as follows. When the abrasive concentration becomes high, the abrasive increases too much between the polishing pad and workpiece, and zone (a) doesn't replace zone (b) any further. As a result, the number of the effective abrasives doesn't increase and MRR becomes constant. [5]

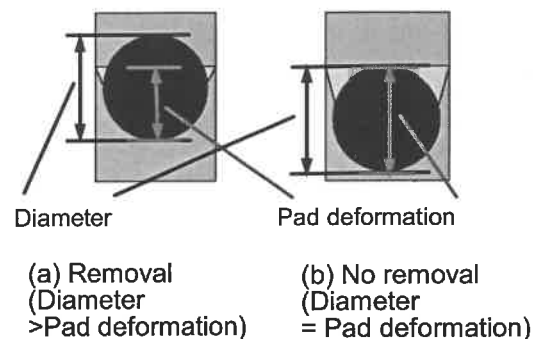


FIGURE 5. Schematic of abrasive – workpiece contact zone.

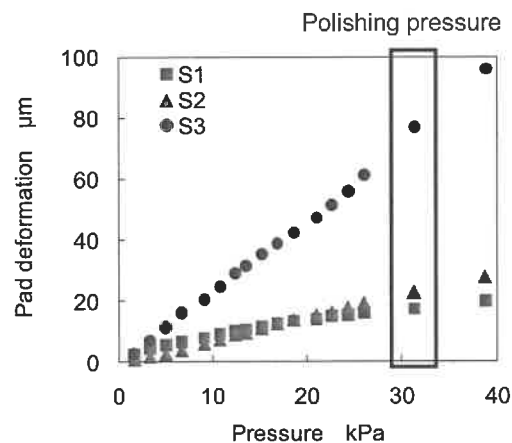


FIGURE 6. Hardness of the polishing pad (S1 > S2 > S3).

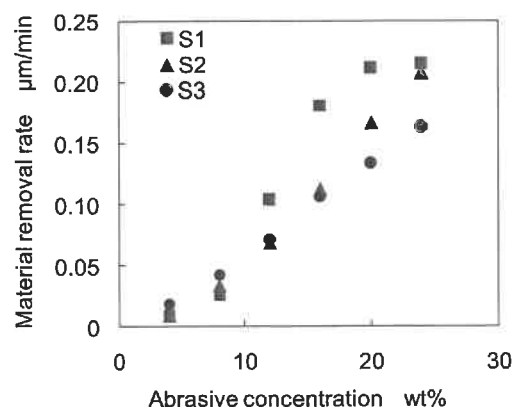


FIGURE 7. Effect of hardness of polishing pad on material removal rate with increase of abrasive concentration.

CONCLUSION

To improve the finishing efficiency, a new polishing model considering contact behavior of polishing pad with workpiece surface was proposed. Based on the model, the efficiency was improved with the usage of hard polishing pad.

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IMPROVEMENT OF TOOL WEAR RESISTANCE IN STEEL CUTTING BY TEXTURED SURFACE

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INTRODUCTION

In the cutting processes, improvement of anti-adhesive properties and wear resistance of cutting tools are strongly required for increasing the tool life. Then, many cutting technologies such as cutting fluid, fluid supply, and cutting tool material and surface coating have been developed. However, as the increase in demand for high-efficiency machining and the decrease of product cost, the requirement for increasing tool life get higher in recent years.

In order to meet these requirements, we adopted surface engineering approach, namely, a functionalization of tool surfaces by textures [1, 2]. In our previous research, we developed a DLC-coated cutting tool with nano/micro-textured surface to determine the role of the textured tool rake face in retaining cutting fluid and decreasing the contact area at the interface between tool surface and chip. Face milling experiments on aluminum alloys showed that the textured surface significantly promoted anti-adhesiveness at the tool-chip interface in wet cutting. Moreover, it was revealed that improved retaining capability of cutting fluid on the tool surface brought a good anti-adhesive property of the developed cutting tool [3, 4].

In this paper, the previously developed tool with nano/micro-textured surface was applied to cutting of steel material (S53C) in the hopes of improving the wear resistance and it was revealed that the serious issues relating to the tool wear still remain. Therefore, the cutting tool with periodical stripe grooves was newly developed to overcome the above-mentioned problem. Moreover, TiAlN coated cutting tool with periodical stripe grooves was also evaluated.

PROBLEMS ASSOCIATED WITH CUTTING TOOL WITH NANO/MICRO-TEXTURED SURFACE

In our previously research, a cutting tool with nano/micro-textured surface was developed by using Femto-second laser technology (Fig. 1) [3, 4]. Firstly, wear resistance of the previously

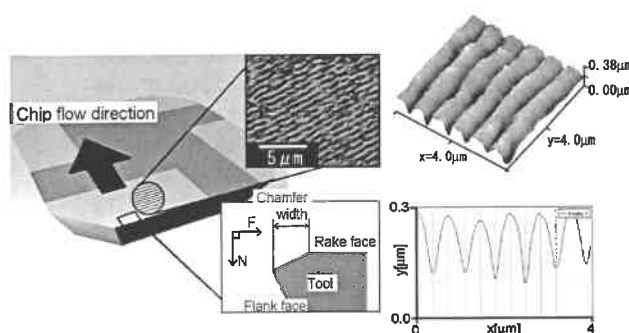


FIGURE 1. Previously developed cutting tool with nano/micro-textured surface [3, 4]

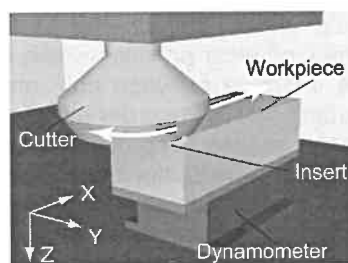


FIGURE 2. Experimental setup of face milling test

TABLE 1. Cutting conditions.

Workpiece	S53C W 60 mm - L 66.4 mm
Tool (Insert)	Cemented carbide K10
Cutting speed	200 m/min (800 rpm)
Depth of cut	2 mm
Feed rate	0.20 mm/rev.
Cutting fluid	Emulsion type (JIS A1) Finecut CFS-100, NEOS
Supply rate	12.6 L/min

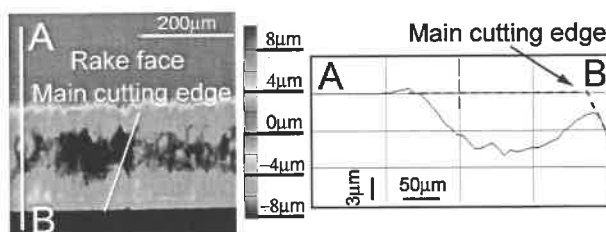


FIGURE 3. Rake face of the cutting tool with nano/micro-textured surface after cutting for 300 m

developed cutting tool in cutting of steel material was evaluated.

Cutting experiments were conducted on steel material (S53C) using a vertical machining center (Yamazaki Mazak Corp., AJV-18), as illustrated in Fig. 2. The center of the cutter was set on the center line of the workpiece. Table 1 lists the cutting conditions. Emulsion type cutting fluid (NEOS Co., Ltd., Finecut CFS-100) was supplied at a flow rate of 12.6 L/min.

Fig. 3 shows the 3-D geometry and cross section of the rake face of the previously developed cutting tool after cutting for 300 m. As shown in this figure, severe crater wear was occurred and the texture on the rake face was completely worn out. From this result, it was found that the nano/micro-textured surface do not have a sufficient wear resistance in cutting of steel material.

PROPOSED CUTTING TOOL WITH MICRO STRIPE GROOVED SURFACE

In order to suppress the tool wear, textured surface have to bring effects such as (a) absorbing stress distribution on the tool rake face and (b) trapping wear particles which cause plowing at the interface between chip and the surface of a cutting tool [5]. In order to absorbing stress distribution, rectangular grooves that have a larger contact area are better than sinusoidal grooves like the nano/micro texture. Moreover, from SEM observation of worn surface of the cutting tool (Fig. 4), the size of wear particle was estimated several micro meters. Therefore, it is considered that the grooves with more than several micro meters wide and deep are required to trap the wear particles. Based on these considerations, a cutting tool with periodical stripe grooves was newly developed (Fig. 5). The regular micro grooves 5 μm deep and 20 μm wide were generated on the rake face of cemented carbide cutting tool using a femto-second laser. In order to evaluate the influence of groove direction, two types of cutting tools with different textured surface, namely, the cutting tool with grooves parallel to main cutting edge and the one with grooves orthogonal to main cutting edge were prepared (see Fig.5).

Fig.5 shows the 3-D geometry and cross section of rake face of the conventional cutting tool and the developed cutting tools after cutting for 300 m. As shown in these figures, it was found that the maximum depth of the crater wear of the parallel groove tool (Fig. 6 (b)) was less than half compared to that of conventional cutting tool

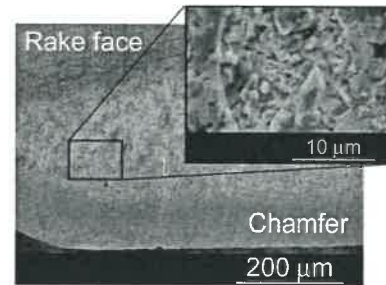


FIGURE 4. Surface of the tool with nano/micro textured surface after cutting

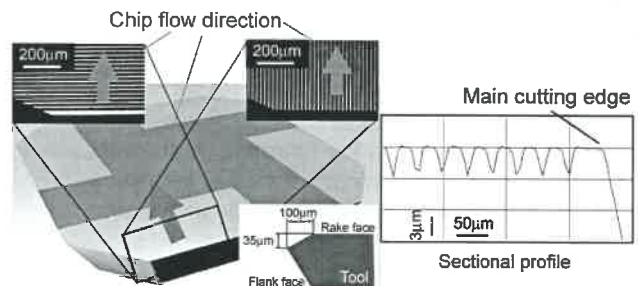
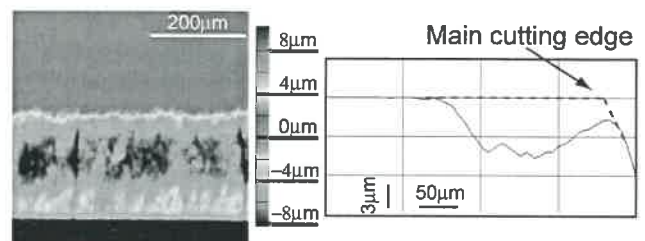
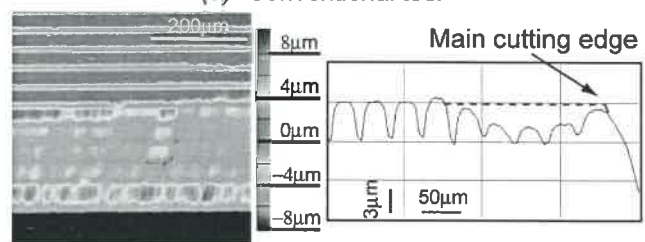


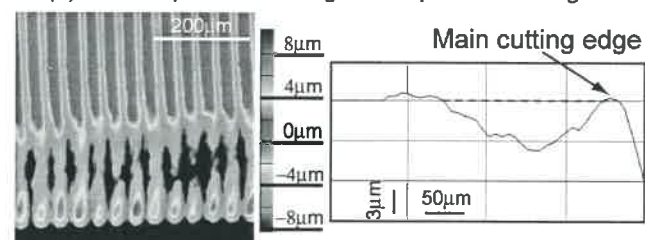
FIGURE 5. Developed cutting tool with micro stripe grooved surface



(a) Conventional tool

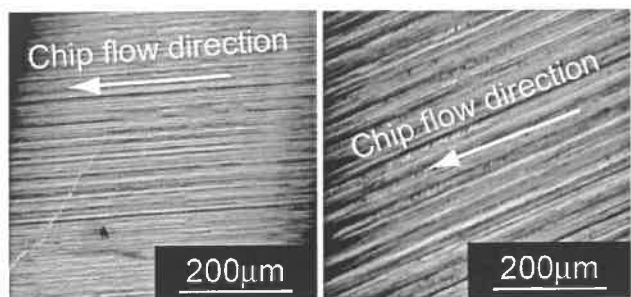


(b) Developed tool with grooves parallel to edge



(c) Developed tool with grooves orthogonal to edge

FIGURE 6. Rake face of the cutting tools after cutting for 300 m



(a) Developed tool with grooves parallel to edge (b) Developed tool with grooves orthogonal to edge

FIGURE 7. Reverse face of chips

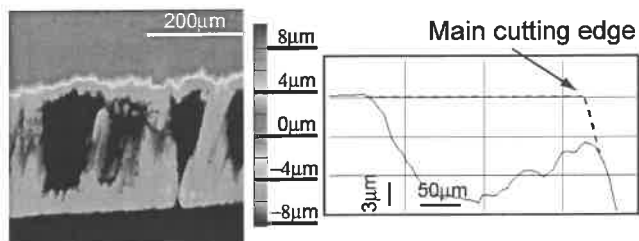
(Fig. 6 (a)). On the other hand, the rake face of the orthogonal groove tool (Fig. 6 (c)) was worn away as in the case of the conventional tool. Figure 7 shows reverse face of removed chips and it was observed that the grooved surface was transferred to reverse face of chips obtained by using the orthogonal groove tool (Fig. 7 (b)). This result indicates that, in the case of the orthogonal groove tool, chips entered into the grooves because the chip flow direction is corresponding to the groove direction and severe crater wear was occurred both in concave and convex area of the texture.

TiAlN COATED CUTTING TOOL WITH MICRO STRIPE GROOVED SURFACE

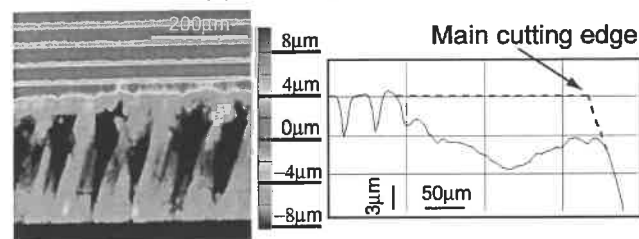
As described above, the cutting tool with periodical stripe grooves parallel to main cutting edge suppressed the crater wear on the rake face. However, the wear resistance of the developed tool was still not sufficient. Fig. 8 shows the rake face of the cutting tools after cutting for 600 m and Fig. 9 shows the changes of the maximum depth of crater wear on the rake face. As shown in these figures, the texture on the rake face was completely worn out at the cutting length 600 m, and the crater wear of the developed cutting tool progressed along with the conventional tool after wearing out the texture on the rake face.

In order to improve wear resistance of the developed cutting tool, TiAlN coated cutting tool with grooved surfaces was developed. After creating the grooves, TiAlN film was coated by Physical Vapor Deposition.

Figure 10 shows the 3-D and 2-D geometry of rake face of the conventional TiAlN coated tool after cutting for 300 m and 600 m. As shown in these figures, chip adhesion and partial wear were occurred in conventional TiAlN coated tool



(a) Conventional tool



(b) Developed tool with grooves parallel to edge

FIGURE 8. Rake face of the cutting tools after cutting for 600 m

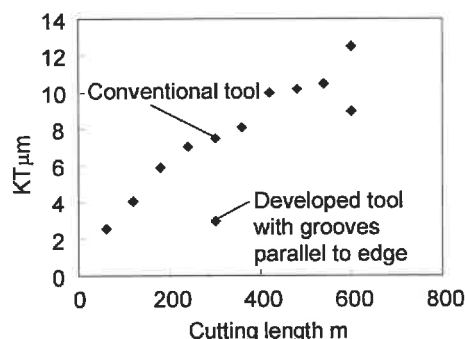
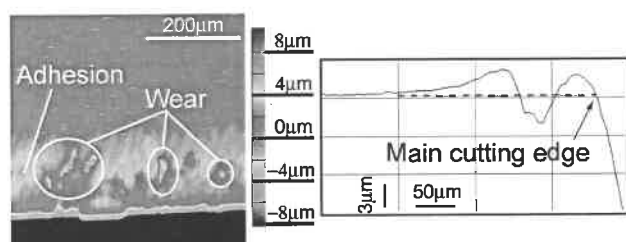
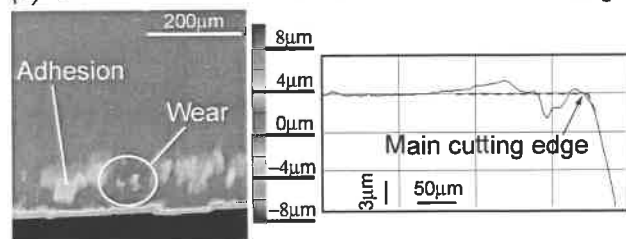


FIGURE 9. Changes of maximum depth of crater wear with cutting length



(a) Conventional TiAlN coated tool after 300 m cutting



(b) Conventional TiAlN coated tool after 600 m cutting

FIGURE 10. Rake face of the TiAlN coated cutting tools after cutting

and they were increased as cutting length increased. Moreover, SEM image and EDX-W, Co analysis of the conventional TiAlN coated tool that indicates the distributions of the W and Co atoms were shown in Fig. 11, and it was found that the tool substrate material (W-Co cemented carbide) was detected at the worn area of the tool rake face. From this result, it was inferred that the TiAlN coating on the tool surface was flaked with abrasion of the chip adhesion.

On the other hand, Fig. 12 shows the tool rake face of the TiAlN coated tool with grooves parallel to the main cutting edge. As shown in these figures, it was found that the textured surface on the rake face significantly suppressed the chip adhesion and prevent TiAlN coating from flaking, although some thermal cracks were observed.

CONCLUSIONS

In order to improve wear resistance of the cutting tool in cutting of steel materials, the cutting tool with periodical stripe grooves was developed, with the following findings:

- Nano/micro-textured surface do not improve wear resistance in cutting of steel materials compared to the conventional tool.
- A cutting tool with periodical stripe grooves was developed and it was found that the developed tool suppressed the crater wear.
- A TiAlN coated cutting tool with grooves parallel to the main cutting edge significantly decrease chip adhesion and prevent TiAlN coating flaking.

ACKNOWLEDGEMENTS

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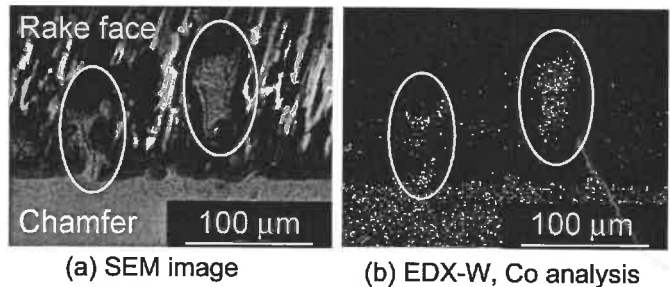
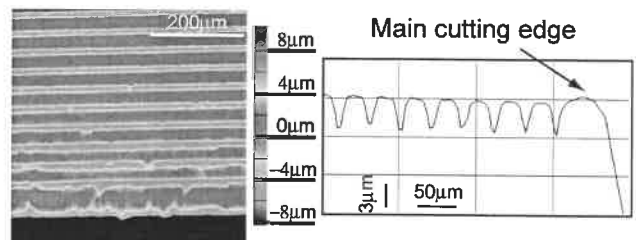
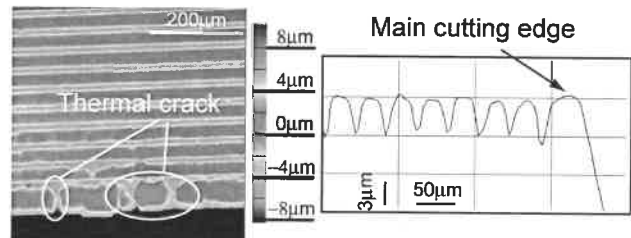


FIGURE 11. Rake face of the conventional TiAlN coated cutting tool after cutting for 600 m



(a) Developed TiAlN coated tool after 300 m cutting



(b) Developed TiAlN coated tool after 600 m cutting

FIGURE 12. Rake face of the TiAlN coated cutting tools after cutting

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ULTRAPRECISION MICROGROOVING ON HARD MATERIAL WITH TOOL WEAR SUPPRESSION

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INTRODUCTION

In recent years, the demand of ultraprecision optic elements is rising with the technical development and miniaturization of information equipments. The molds, which are used in manufacturing optic elements, need to be fabricated with high efficiency and accuracy. Plastic is used as the material of optic elements by the advantage of low cost and low molded temperature. However, it is requested to use glass as the material of optic equipments because of its excellent optical property. Thus, the molds are requested to resist high temperature. As the material with heat-resistance and hardness, hard material including cemented carbide or ceramics is used. When these hard materials are machined, severe tool wear occurs, which causes deterioration in the accuracy of surface.

A machining method which has the ability to suppress tool wear, called cutting point swivel machining [1], is proposed. This study aims at making clear the mechanism of the cutting point swivel machining, and at investigating the relationship between the rotation speed of tool and the tool wear.

ULTRAPRECISION MACHINING CENTER

An ultraprecision 5-axis machining center (FANUC ROBOnanoUi) is used in the machining experiments. The machining center is composed of three translation axes, X, Y and Z, and two rotation axes, B and C. The resolution of positioning accuracy of each translation axis is 1 nm, and that of rotation axis is 0.00001 deg.

The non-rotational cutting tool (A.L.M.T DIAMOND Corp. Ultra Precision Cutting Tool) used in the experiment is made of a single crystal diamond with the radius of 0.1 mm. As shown in Fig.1, there is a special chamfer on the rake face [2]. The width of chamfer is 2 to 4 μm . The angle of chamfer is set to be -30 deg.

CUTTING POINT SWIVEL MACHINING

In the conventional microgrooving, the posture of tool is not changed to make good surface roughness. However, as only a part of tool tip is used in the machining, tool wear converges on this part. To solve this problem, a method called "Cutting point swivel machining" is proposed. As shown in Fig.2, the tool is swiveled in the plane perpendicular to the tool feed direction during machining. Cutting point is always changing in the machining so that the temperature of tool tip is not increased. In addition, tool wear can be averaged by using all part of tool tip. Thus, it is expected that the tool wear can be suppressed by using cutting point swivel machining.

By the verification experiment of machining SiC, the machining distance of cutting point swivel machining is twice as long as that of the conventional microgrooving, while tool wear being suppressed substantially [1].

MECHANISM OF CUTTING POINT SWIVEL MACHINING

The mechanism of cutting point swivel machining is investigated. Figure 3 shows SEM photos of machined microgrooves. (a) shows the microgroove machined by conventional machining. (b) and (c) show the microgrooves machined by cutting point swivel machining with the rotation speed of 16 and 286.5 deg/min, respectively. The part between white dashed lines is the surface of microgroove. It is confirmed that all microgrooves

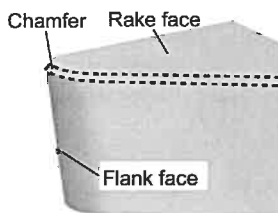


Fig.1 Tool with special chamfer

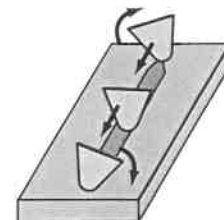


Fig.2 Cutting point swivel machining

are machined in ductile mode because the cutting mark can be seen. The depth of cut is set to be 80 nm.

By using the cutting point swivel machining, when the rotation speed is 16 deg/min, the cutting marks are almost paralleled with the feed direction, which is the same as the conventional microgrooving. To compare with this, when the rotation speed is 286.5 deg/min, the cutting mark is inclined with the feed direction, and the angle between cutting mark and feed direction is about 26 deg. Let's consider the reason of this phenomenon. As shown in Fig.4, the tool is fed, while being rotated by using the cutting point swivel machining, so the workpiece is subjected to the force both in the tool feed direction and in the tool rotation direction at the same time. The direction of the chip ejection is different from

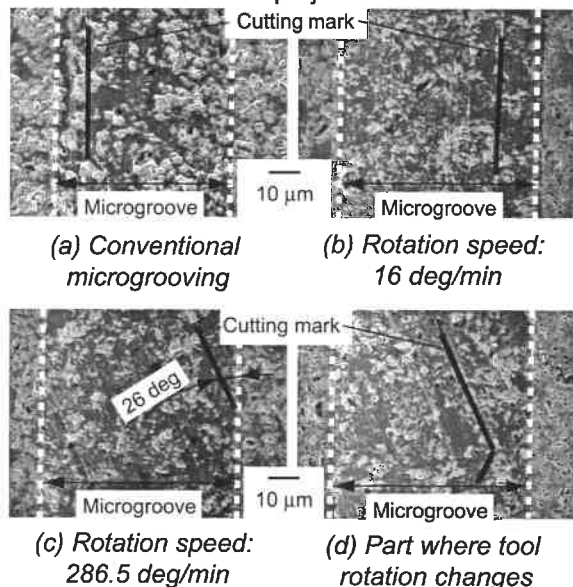


Fig.3 Machined microgrooves

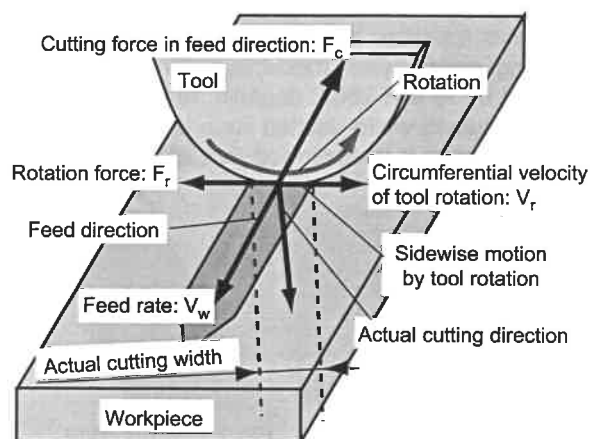


Fig.4 Mechanism of cutting point swivel machining

the tool feed direction, and the actual cutting direction is inclined. When the tool rotation speed is 16 deg/min, to compare with the feed rate, the rotation speed is too slow, so the cutting mark is almost parallel to the feed direction.

The ratio r_v of the circumferential velocity V_r to the feed rate V_w is calculated by the tool rotation speed and the radius of tool. Feed rate V_w is 1 mm/min, the radius of tool is 0.1 mm. When the tool rotation speed is set to be 16 and 286.5 deg/min, the r_v are 0.028 and 0.5, respectively. By this ratio r_v , the degree between actual cutting direction and tool feed direction can be calculated as 1.604 and 26.565 deg, respectively. The actual cutting direction is the same as the cutting mark. It is confirmed that the use of the cutting point swivel machining changes the actual cutting direction along with the rotation of tool.

When the tool rotating direction is changed from plus to minus direction of C axis, there is the threat of deterioration of the machined surface because of the back rush of the rotation axis. Fig.5(d) shows the part where the tool rotating direction is changed. It is seen that the tool rotation direction is changed because the direction of cutting mark is changed. As shown in this figure, the good microgroove shape can be obtained even in case that the tool rotating direction is changed.

ROTATION SPEED AND TOOL WEAR

As shown in Fig.4, by increasing the tool rotation speed, the moving distance of the tool tip becomes longer to remove the same volume of workpiece. The actual width of cut becomes smaller, and the cutting resistance in the tool feed direction can be reduced [3]. Then, the cutting fluid is easily supplied between the tool tip and the workpiece by accelerating the tool rotation speed, to suppress tool wear.

An experiment is conducted to investigate the relationship between tool rotation speed and tool wear[4]. The tool rotation speed is set to be 6, 12,

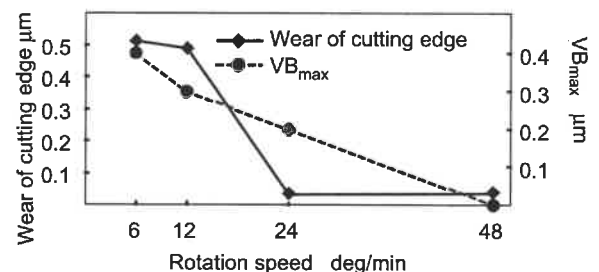


Fig.5 Tool wear of the rotation speed

24 and 48 deg/min. The result is shown in Fig.5. The solid line shows the wear of cutting edge which can be calculated by the width of chamfer before and after machining, and the dashed line shows the maximum tool wear on flank face, VB_{max} . As shown in this figure, when the rotation speed is increased from 6 to 48 deg/min, both the tool wear of cutting edge and the maximum tool wear of flank face can be suppressed.

Then, the cutting forces are estimated by use of parameters; the circumferential velocity of the tool rotation V_r , the feed rate V_w , the cutting force in the tool feed direction F_c , and the rotation force F_r , which is the resistance generated by the rotation of tool. The cutting mode is set so as to minimize the power of the cutting which is shown in Eq.(1) [5].

$$P = F_c V_w + F_r V_r \quad (1)$$

The estimated cutting forces are shown in Table 1[6]. It is confirmed that when the rotation speed of tool is increased from 6 to 48 deg/min, the reduction in cutting force is too small to suppress tool wear. As a result, it is presumed that the tool wear is reduced due to the improvement of lubrication effect and the cooling effect of cutting fluid.

However, as listed in Table 1, when the rotation speed of tool becomes larger, the cutting force is expected to become small substantially.

TABLE 1 Estimated results

Tool rotation speed	r_v	$F_c / A\tau_s$	$F_r / A\tau_s$	$F_v / A\tau_s$
6 deg/min	0.003	355.98	0.01	911.16
12 deg/min	0.007	355.97	0.03	911.12
24 deg/min	0.014	355.91	0.06	910.96
48 deg/min	0.028	355.65	0.12	910.32
	0.100	351.84	0.42	900.55
	0.500	282.12	1.67	722.10
	1.000	193.76	2.29	495.92

TABLE 2 Cutting conditions

Workpiece	SiC
Total depth	5 μ m
Depth of cut	80 nm
Rotation speed	16 and 286.5 deg/min
Feed rate	1 mm/min
r_v	0.028 and 0.5
Cutting fluid	Oil mist

EXPERIMENT WITH HIGH ROTATION SPEED

In this experiment, the ratio of the circumferential velocity to the feed rate, r_v , is set to be 0.028 and 0.5. The limit of synchronized feed rate of the machining center is 720 min^{-1} , so the feed rate is set to be 1 mm/min, and the tool rotation speed is 16 and 286.5 deg/min, respectively. The cutting condition is shown in Table 2. The machining shape is microgroove with length of 12.5 mm and depth of 5 μ m. 15 microgrooves are machined by each rotation speed, and the machining distance is 11,250 mm.

Figure 6(a) shows the rake face of the tool with the rotation speed of 16 deg/min before and after machining. The width of chamfer is 3.908 μ m before machining, and it reduces to 3.531 μ m. The tool wear of cutting edge is about 377 nm. Then, several micro chippings can be seen at the cutting edge of tool.

Then, as shown in Fig.6(b), when the rotation speed is 286.5 deg/min, the width of chamfer before and after machining is 3.865 μ m and 3.724 μ m. The tool wear of cutting edge is about 100 nm. By comparing the tool with the rotation speed of 16 deg/min, it is found that the tool wear of cutting edge is reduced. However, there are more micro chippings on the cutting edge.

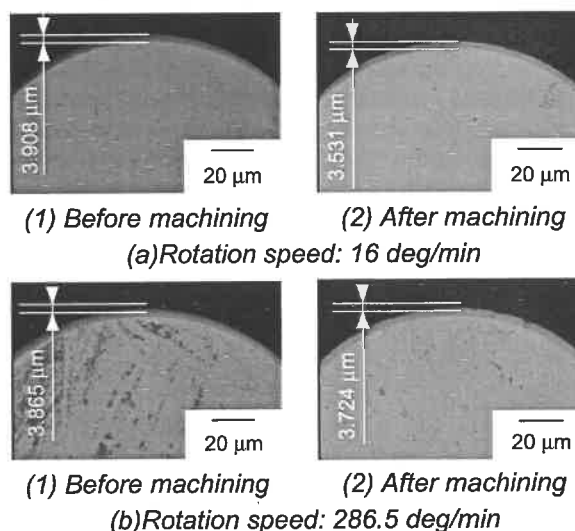


Fig.6 Rake face of the tool used in the machining

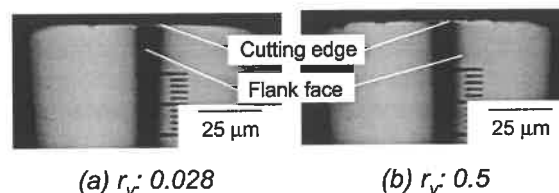


Fig.7 Flank face of the tool used in the machining

The flank face of the tool is shown in Fig.7. The maximum tool wear on flank face, $VB_{\max}$ is 0.7 and 0.6 μm respectively. The maximum width of the micro chipping on the flank face is 1.8 and 2.8 μm , respectively. The number of micro chipping with the rotation speed of 286.5 deg/min is more than that with the rotation speed of 16 deg/min.

It is confirmed that by increasing the tool rotation speed, both tool wear of cutting edge and tool wear on the flank face can be reduced. The reason is that cutting forces can be reduced by increasing tool rotation speed. Then, the reason that both number and size of the chippings are increased by accelerating the tool rotation speed is investigated.

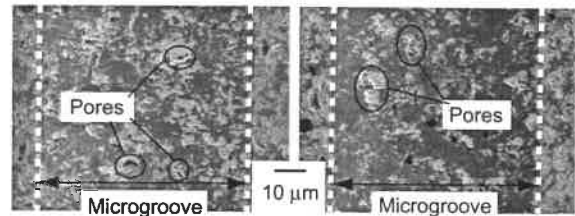
Figure 8 shows the machined microgrooves. Microgroove machined by each rotation speed shows good shape accuracy. However, because poreless processing of the workpiece is not executed, there are large numbers of pores in the workpiece. The cutting forces are changed by the presence or absence of pores. When the tool rotation speed is increased, the impact to the tool becomes large, and it may cause the chipping on the cutting edge.

Then, the transition in the roughness of machined microgrooves is shown in Fig.9. When the rotation speed is 16 deg/min, good roughness can be obtained until the 12th microgroove. However, when the rotation speed is 286.5 deg/min, the roughness becomes worse from the 7th microgroove. The reason is the occurrence of the chipping on the cutting edge. The chipping occurs in the 7th microgroove, and it becomes larger and larger in the machining to cause the deterioration of the microgroove roughness.

CONCLUSION

This study aims at investigating the mechanism of the cutting point swivel machining and the relationship between the tool rotation speed and tool wear. As a result, the conclusion can be summarized as follows:

1. By use of cutting point swivel machining, it is confirmed that the actual cutting direction can be changed by the rotation of the tool.
2. Cutting forces can be reduced with increasing the tool rotation speed. However, the chipping on cutting edge is increased since the damage of the tool becomes larger at the same time. By this reason, the roughness of machined microgrooves becomes worse.



(a) Rotation speed: 16 deg/min (b) Rotation speed: 286.5 deg/min

Fig.8 Machined microgrooves

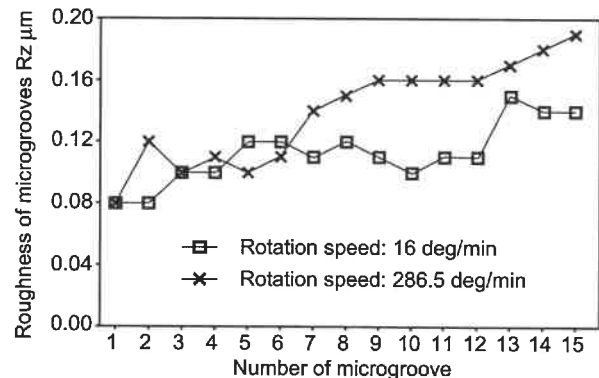


Fig.9 Measured surface roughness Rz

ACKNOWLEDGMENT

The authors would like to express their sincere gratitude to FANUC LTD for donating ROBOnanoUi and to Nippon Steel Materials Co., Ltd for supplying workpieces.

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INTERNAL LASER MARKING FOR HIGHER RETAINED STRENGTH

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ABSTRACT

A novel internal laser marking technique has been developed for glass display production. Internal laser marking, compared to mechanical scribing, chemical etching and additive printing, is cleaner and easier to be integrated into clean room processes. The developed laser marking system consists of a frequency-tripled diode-pumped solid-state laser with a wavelength of 355 nm, a laser beam expander, a galvanometer based beam steering system to generate any 2D features, a marking control interface and the mechanical body including a working table and several chucks to hold glass substrates. It is confirmed by our experiments that the internal laser marking technique is able to retain ~80% of glass ROR (ring-on-ring) strength under carefully controlled laser parameters, especially the focus Z-location.

INTRODUCTION

The increasing use of thin display glass in lightweight, portable handheld electronics, computer and mobile phones has presented significant challenges to every individual process because

of the sensitivity of glass strength to surface and subsurface flaws. Marking, as a key identification step to trace product quality and history, is also required to have higher retained strength so as to avoid serious glass breakage in the whole process of production.

Currently, mechanical scribing, additive printing, laser marking, and chemical etching [1] are the most often used and commercially available marking techniques. Mechanical scribing is mostly used for metal and plastic parts, and it usually creates a damage layer of around 30 μm if it is used to mark glass parts and causes a significant drop in glass breakage resistance. Additive printing does not cause any damage to glass, because it is only adding a film of organic or carbonate with the thickness of ~100 μm on the surface of glass parts. Unfortunately, these additive marks are, likely, not compatible to clean-room processes. Chemical etching is able to, by its nature, produce indelible marks on glass; however, it is a rather complex process and not environmentally benign.

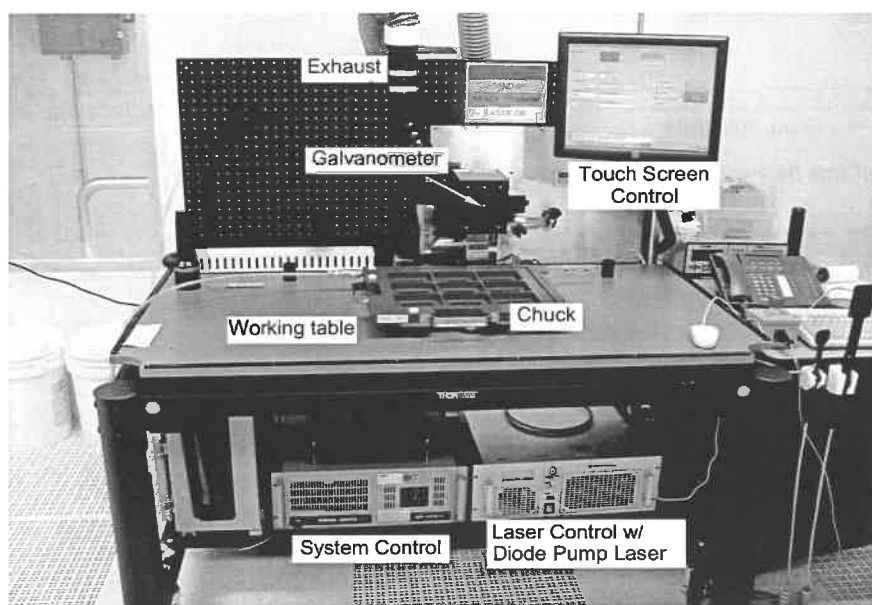


Fig.1 View of the laser marking system

Laser marking has a number of advantages over its peers [2]. Laser marking employs a focused laser beam to ablate a certain layer of glass away or modify it into visible damages (marks). It has most of the attributes required by the display production, such as high marking speed, low operating cost, and contamination-free. This paper is focused on process development and process characterization.

LASER MARKING PRINCIPLE & SYSTEM

Material removal in laser marking of glass is also based on the mechanism of laser ablation, i.e. the combination of glass melting, vaporizing, and modifying. The developed laser marking system consists of a frequency-tripled diode-pumped solid-state laser with a wavelength of 355 nm, a laser beam expander, a galvanometer based beam steering system to generate any 2D features, a marking control interface and the mechanical body including a working table and several chucks to hold glass substrates. Fig.1 shows a view of the laser marking system.

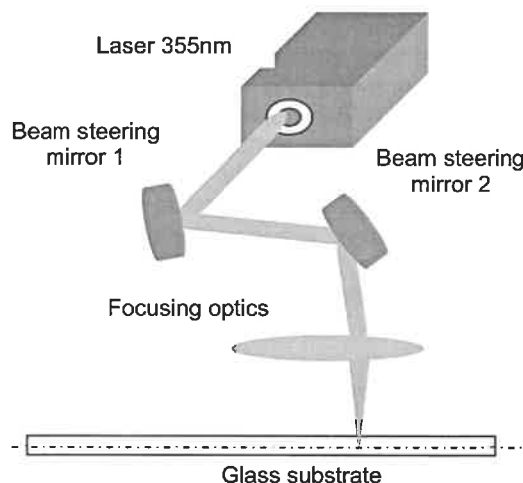


Fig.2 View of the beam steering system

The galvanometer based beam steering system controls the laser beam to generate various 2D characters, e.g. numbers, letters and symbols. Fig.2 gives a schematic view of the steering system. The 355nm laser beam is first expanded by using the beam expander, and then guided to the 2 galvanometer driven mirrors, and finally focused by a special 56 mm f-theta lens to the required focus spot size and the required focus depth. For internal marking, both the beam diameter expansion and the short 56 mm focal length of the lens are required to achieve a tight focus and small depth of focus. If the depth of

focus is too large the damage (mark) can not be sufficiently contained within the glass.

LASER MARKING PROCESS ANALYSIS

Reasonable visibility of identification marks is another metric for a successful marking, apart from glass strength drop. Increasing the size of marks (more accurately speaking, the features of marks) is an efficient way to improve the visibility of marks. However, if the marks are created by material removal, an increase in the size of marks will cause a decrease in fracture strength of marked parts, as shown in Fig.3.

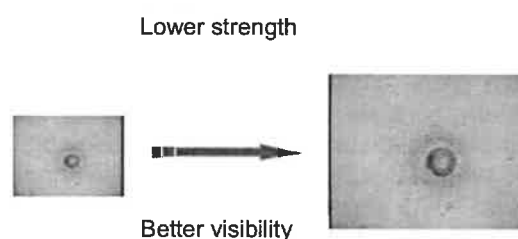
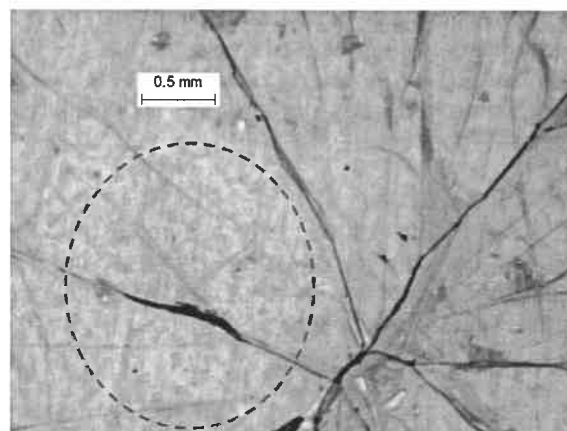
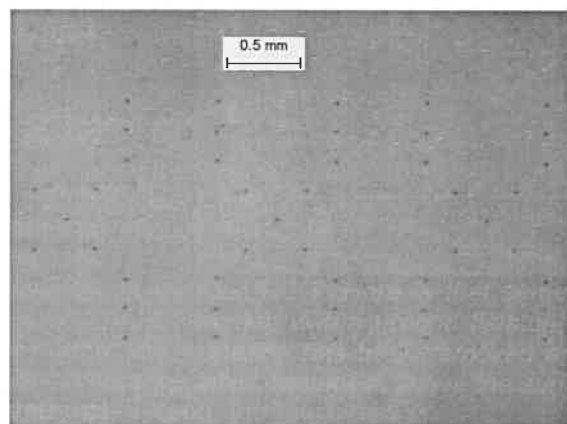


Fig.3 Visibility vs. fracture strength



(a) Continuous scanning



(b) Dot matrix

Fig.4 Laser marks: continuous and dot matrix

24 and 48 deg/min. The result is shown in Fig.5. The solid line shows the wear of cutting edge which can be calculated by the width of chamfer before and after machining, and the dashed line shows the maximum tool wear on flank face, VB_{max} . As shown in this figure, when the rotation speed is increased from 6 to 48 deg/min, both the tool wear of cutting edge and the maximum tool wear of flank face can be suppressed.

Then, the cutting forces are estimated by use of parameters; the circumferential velocity of the tool rotation V_r , the feed rate V_w , the cutting force in the tool feed direction F_c , and the rotation force F_r , which is the resistance generated by the rotation of tool. The cutting mode is set so as to minimize the power of the cutting which is shown in Eq.(1) [5].

$$P = F_c V_w + F_r V_r \quad (1)$$

The estimated cutting forces are shown in Table 1[6]. It is confirmed that when the rotation speed of tool is increased from 6 to 48 deg/min, the reduction in cutting force is too small to suppress tool wear. As a result, it is presumed that the tool wear is reduced due to the improvement of lubrication effect and the cooling effect of cutting fluid.

However, as listed in Table 1, when the rotation speed of tool becomes larger, the cutting force is expected to become small substantially.

TABLE 1 Estimated results

Tool rotation speed	r_v	$F_c / A \tau_s$	$F_r / A \tau_s$	$F_v / A \tau_s$
6 deg/min	0.003	355.98	0.01	911.16
12 deg/min	0.007	355.97	0.03	911.12
24 deg/min	0.014	355.91	0.06	910.96
48 deg/min	0.028	355.65	0.12	910.32
	0.100	351.84	0.42	900.55
	0.500	282.12	1.67	722.10
	1.000	193.76	2.29	495.92

TABLE 2 Cutting conditions

Workpiece	SiC
Total depth	5 μ m
Depth of cut	80 nm
Rotation speed	16 and 286.5 deg/min
Feed rate	1 mm/min
r_v	0.028 and 0.5
Cutting fluid	Oil mist

EXPERIMENT WITH HIGH ROTATION SPEED

In this experiment, the ratio of the circumferential velocity to the feed rate, r_v , is set to be 0.028 and 0.5. The limit of synchronized feed rate of the machining center is 720 min^{-1} , so the feed rate is set to be 1 mm/min, and the tool rotation speed is 16 and 286.5 deg/min, respectively. The cutting condition is shown in Table 2. The machining shape is microgroove with length of 12.5 mm and depth of 5 μ m. 15 microgrooves are machined by each rotation speed, and the machining distance is 11,250 mm.

Figure 6(a) shows the rake face of the tool with the rotation speed of 16 deg/min before and after machining. The width of chamfer is 3.908 μ m before machining, and it reduces to 3.531 μ m. The tool wear of cutting edge is about 377 nm. Then, several micro chippings can be seen at the cutting edge of tool.

Then, as shown in Fig.6(b), when the rotation speed is 286.5 deg/min, the width of chamfer before and after machining is 3.865 μ m and 3.724 μ m. The tool wear of cutting edge is about 100 nm. By comparing the tool with the rotation speed of 16 deg/min, it is found that the tool wear of cutting edge is reduced. However, there are more micro chippings on the cutting edge.

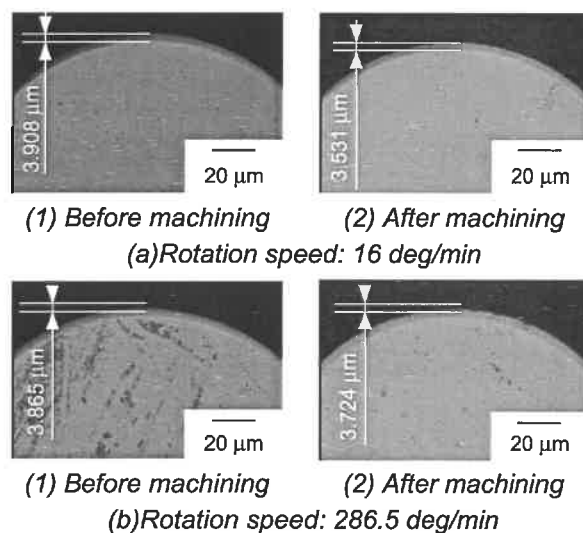


Fig.6 Rake face of the tool used in the machining

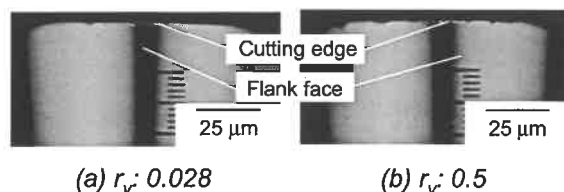


Fig.7 Flank face of the tool used in the machining

Fig.4 gives a view of laser marks generated by 2 different recipes, dot matrix and continuous. The laser marks made by the continuous scanning are visible with our naked eyes, while the ones in dot matrix are only visible with the help of back lighting. ROR (ring on ring) strength tests were also conducted with the samples made in these experiments. Fig.5 summarizes the ROR strength data of 3 separate tests. In Test 1 and Test 2, the continuous scanning recipe was used, while in Test 3 the dot matrix recipe was adopted. In each ROR test, a group of controls, the ROR strength of non-marked glass samples, are added to rule out the effect of test operating differences. As shown in Fig.5, the ROR strength of laser marked samples is significantly lower than the control, although the dot matrix samples are a bit better than the continuously marked ones. For continuous scanning, the retained ROR strength is about 103 MPa, while for dot matrix it is ~ 206.6 MPa.

Experiments were also conducted to investigate the effect of mark locations on the retained glass strength, in which 4 mark locations were tested, like 0, 25, 75, and 150 μm from the surface. 0.55 mm thick Corning glass is used here.

Fig.6 shows the ROR strength data of these samples with 4 different mark locations. It is

clearly seen that the mark location has a significant impact on the ROR fracture strength. When the distance from the marks to the back surface is 0 or 25 μm , the marks are similar to surface flaws and bring the ROR strength down to around 70 MPa. When the marks are made below the surface around 75 μm , their effects on the ROR strength reduce and The ROR strength become around 260 MPa, even though the marks are of the same size as the earlier ones. If the marks are adjusted to 150 μm below the back surface, their effects on the ROR strength become even less. This group of samples has its ROR fracture strength of about 430 MPa.

SUMMARY

Internal laser marking was developed for thin glass display production. The dot matrix recipe showed a smaller ROR strength drop, compared to the continuous scanning. The mark location was identified having the most significant impact on fracture strength, which can bring it from around 400 Mpa down to 70 MPa.

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MICROFINISHING PRECISION-GROUND SURFACES USING FLATSTONES

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ABSTRACT

Flatstones are used by Professional Instruments to ensure that a work piece will set precisely upon a mating surface. They are essential to maintaining a high level of accuracy, precision and repeatability. A flatstone removes swellings that occur from nicks and dents as well as other imperfections that can occur on an otherwise flat surface. Not only does a flatstone mitigate these imperfections, it can also improve the finish of a precision-ground surface. This article will discuss the effects and benefits of using such flatstones to microfinish precision-ground surfaces.



FIGURE 1. A pair of flatstones being used to prepare a surface plate for a workpiece.

HISTORY

Flatstones have been used at Professional Instruments for over 40 years. The first flatstones were made by Harold Arneson and he ground them to exceptional flatness. Since then the manufacturing process has become more efficient but the basic principle has stayed the same.



FIGURE 2. Flatstones being roto-ground at Professional Instruments

FLATSTONES TODAY

The stones are made out of Aluminum Oxide, the same material used in knife sharpening stones. They are ground flat on both faces with a diamond wheel. The long sides are ground perpendicular to allow stoning up to a shoulder and the ends are unground. Each surface is ground flat to a micron or better. Flatness is crucial to avoid scratching and the cavities in the material are necessary for cutting off local high material. Occasionally, we make stones in special sizes or shapes such as a taper or a radius for an inner diameter surface.

MICROFINISHING USING A FLATSTONE

When a flatstone is rubbed on a surface, small imperfections and high spots are caught in the porous cavities of the flatstone and are sheared off. As the sharp edges around the high spot are trimmed by the flatstone, the area that is in contact between the stone and the part becomes relatively large and the cutting action essentially stops. This idea of shearing high spots can also apply to the finish of a piece. In addition to preparing or repairing a piece, a flatstone can be used to improve the surface finish. In grinding there are inevitable variations in the finish which occur due to many factors such as feed rate, spindle speed and machine tool stiffness. A flatstone can help reduce the effects of these variations, by blunting the highest peaks. In other words, a flatstone can microfinish a precision-ground surface.

THE TEST

We ground several pieces of mild steel, each to a different surface finish ranging from 5 to 13 μin . This was done on a Mitsui CNC Surface Grinder with a roto-grinding attachment. We scanned the surfaces before and after stoning using a Mitutoyo SJ-400 profilometer with a diamond stylus.

Below is an example of a reading before and after stoning.

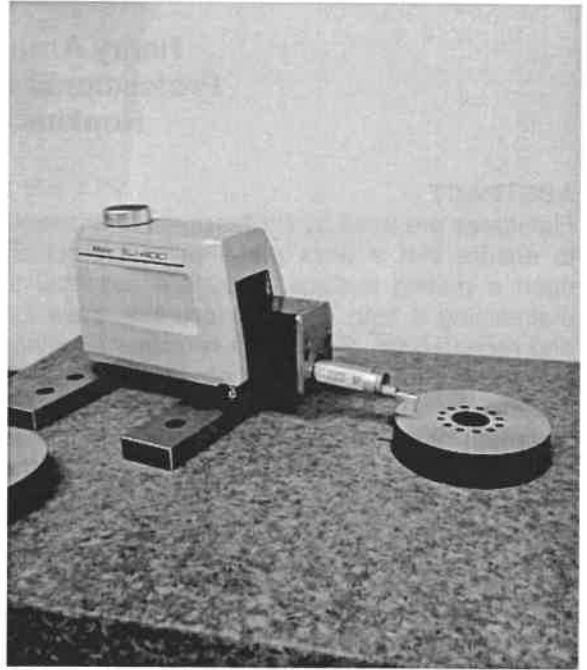


FIGURE 3. Mitutoyo Profilometer

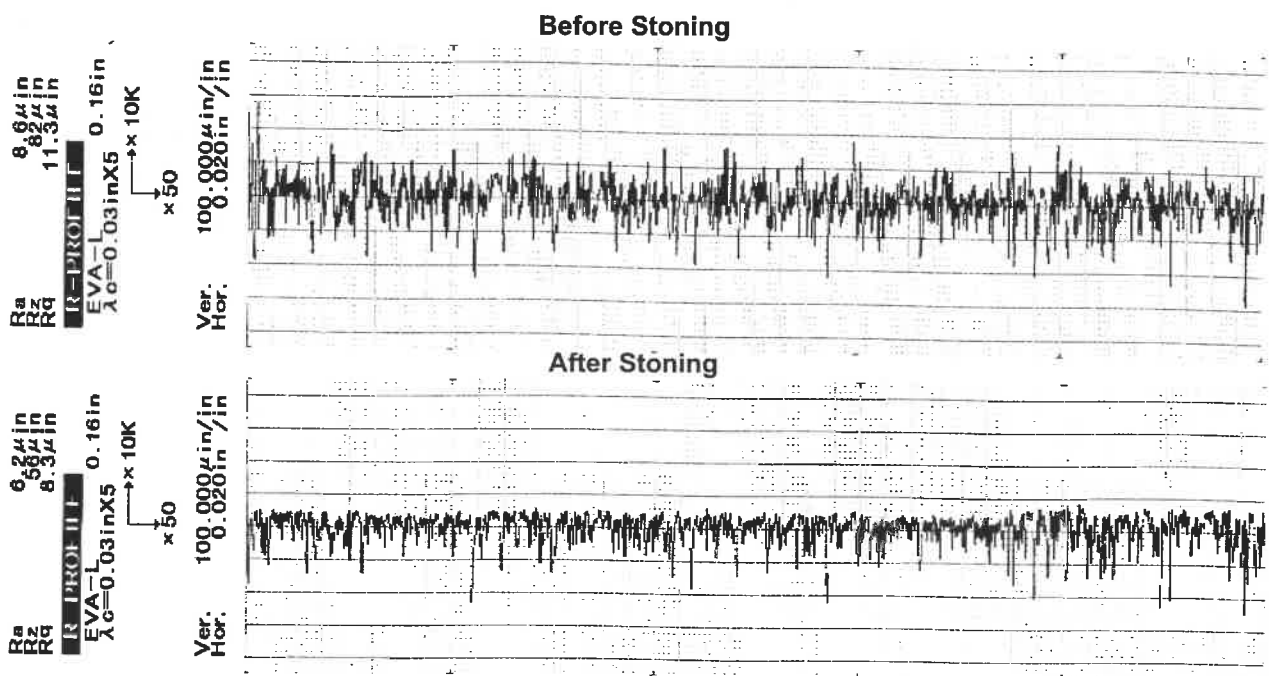


FIGURE 4. Profilometer Readings before and after stoning. Note how in the second reading the irregularities have been "sheared" off, resulting in a flatter surface. The finish indicated by R_a (Roughness Average) started at $8.6 \mu\text{in}$ and after stoning improved to $6.2 \mu\text{in}$.

Table 1. Profilometer Readings taking in Ra (Roughness Average)

Before (μin)	After (μin)	Percent Improvement
11.9	9.6	19%
9	6.4	29%
11.4	8.6	25%
11.1	6.7	40%
11	8.5	23%
8.6	6.2	28%
7.8	6	23%
7.9	5	37%
5.4	4.7	13%
6.8	3.7	46%

Above are readings from different samples prior to and after 1 minute of rigorous stoning. Depending on how hard the stone is applied the percent improvement ranges from 19.33 to even 45.59 percent improvement in the surface finish. These readings are based off of the Roughness Average, as explained by this poster clipping below taking from Surfcom: Surface Finish Parameters provided by Zeiss.

Ra Roughness Average / Arithmetic Average (ISO 468, ANSI B46.1, DIN 4768, JIS B0601)

Arithmetic average of the deviation of the peaks and the valleys on the profile from the mean line. Also called CLA, Center Line Average.

$$Ra = \frac{1}{L} \int_0^L |f(x)| dx$$

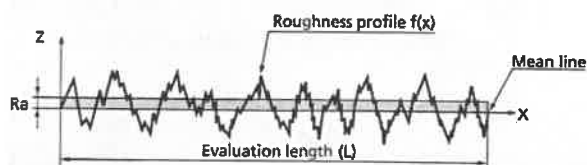


Figure 5. Excerpt from a poster describing the formula for determining the 'Roughness Average.'

There is also a visible change to the surface. The high areas in an otherwise flat surface as they get sheared off as the stone bears down on the surface take on a mirrorlike quality.

SURFACE REPAIR USING FLATSTONES

Flatstones can also be used to repair a damaged surface, as Eric Marsh's paper, "Repairing a Damaged Part with Flat Stones" suggests. In this study a part was deliberately

damaged and then repaired with a flatstone. Here we have conducted our own experiments to repair a part with a flatstone after it has been damaged in similar ways.

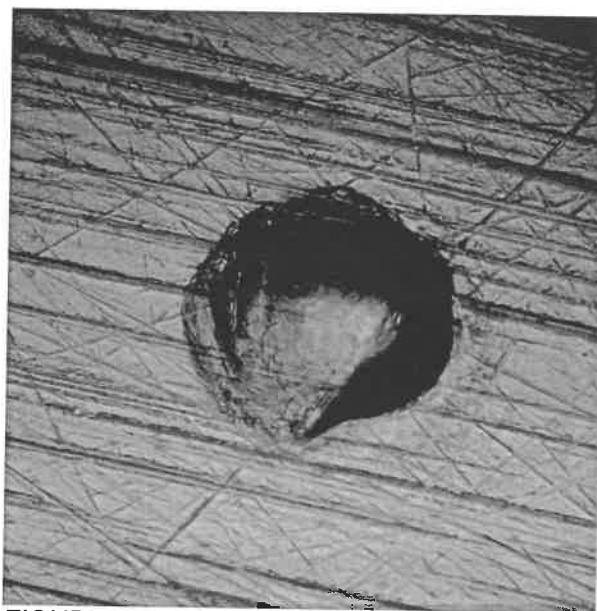


FIGURE 6. Here is a dent in a precision ground surface using a machinist's punch. Notice the swelling that occurs around the dent.

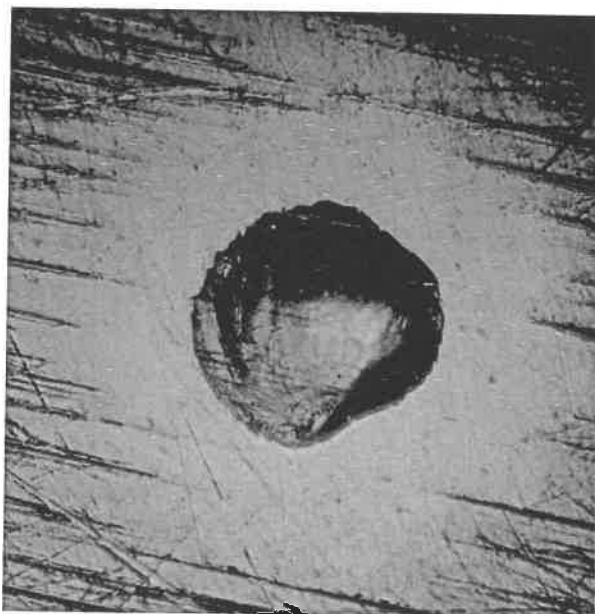


FIGURE 7. Here is that same dent after being treated with a ground flatstone. Notice how the swelling has been removed and no part of the initial damage is above the surface of the part.

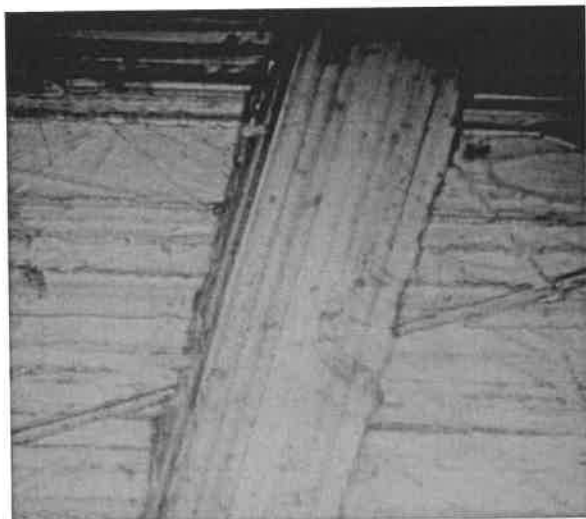


FIGURE 8. Here is a scratch in a precision ground surface using a carbide scratching implement. The displaced material creates parallel ridges on both sides of the furrow.

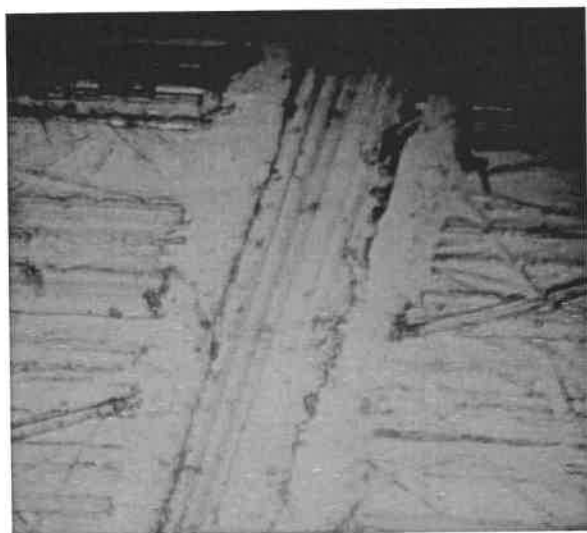


FIGURE 9. Here is the same scratch after stoning. Notice that the ridges have been leveled.

It is evident that the use of the flatstone has removed some of the high areas on these surfaces.

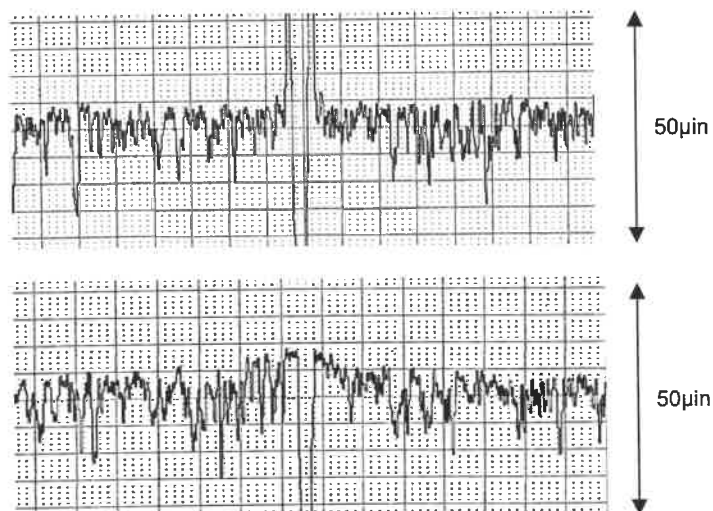


Figure 10. Profilometer scans of a scratch made in a ground surface. Notice the raised edges on either sides of the valley created in the first scan and notice how they disappear after stoning in the second scan.

CONCLUSION

Flatstones are not only essential components to precision assembly; they also can be used to improve the surface quality of a precision ground surface.

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BALANCING AIR BEARING SPINDLES FOR SERVO TRACK WRITERS

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ABSTRACT

This abstract details the importance of balancing spindles especially when used for servo track writing on hard disk drives (HDDs). The demand for data has prompted the makers of HDDs to write higher density and smaller tracks, to accommodate more bytes of information per inch. Therefore, modern day servo track writing requires spindles of extremely small tolerances, and even slight vibration can interfere with the process. Since unbalance can cause unwanted vibration Professional Instruments takes great care to balance each air bearing spindle.

INTRODUCTION

The driver for increases in storage density is customer demand for data storage. According to YouTube's official blog, 48 hours of video are uploaded to their site every minute [1]. This translates to approximately 600 Gigabytes of additional data being stored just on YouTube every minute. In response to this huge demand, the price of data storage on hard drives has decreased dramatically over the last thirty years (Figure 1).

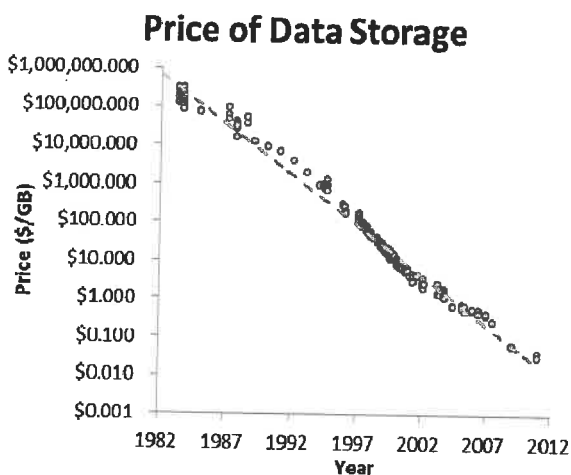


FIGURE 1. The price of gigabytes from 1982-2011 [2]. The dollar values are shown on a logarithmic scale to be able to see both the enormous change at the beginning and the continuing change the last few years.

Manufacturers of HDDs have been able to increase the quantity of data storage while decreasing the price by increasing the density of data, or areal density on disks. Areal density is the product of track density and linear density, or bit density. In the last decade, track density has increased faster than linear (Figure 2) because of better read/write head designs, and improved servo writing technologies [3].

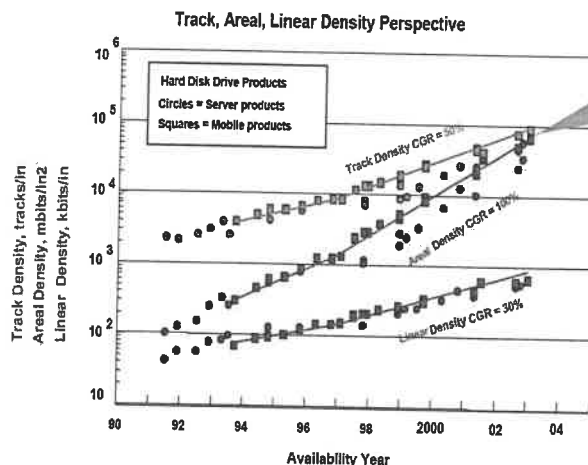


FIGURE 2. Graph representing the increase of track density, linear density, and areal density over time [3].

On hard disks, bytes of data are organized into tracks (concentric circles or spirals) and sectors (wedge divisions), shown in Figure 3 [4]. As track density increases, spiral instead of circular tracks have some advantages, but both types are in use today. By increasing the number of tracks on a disk, you increase the possible data storage.



FIGURE 3. Not-to-scale diagram of tracks and sectors. Present HDDs tracks are much smaller.

The first HDD created by IBM (Figure 4) was as large as a refrigerator but still could only hold 5 Megabytes. The early technology wrote tracks that were 0.05 inches wide, writing only 20 tracks per inch [5].

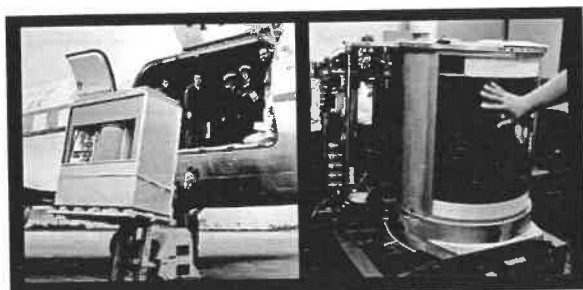


FIGURE 4. The first hard disk drive, Model 350 Disk File, was made by IBM in 1955 and held less than 5 MB. [1]

Today, companies are working on writing 300,000 tracks per inch on HDDs to make compact HDDs that can hold a full terabyte (one million Megabytes) or more. This allows our computers to hold more photos, videos, and music, to run faster and more reliably.



FIGURE 5. The latest HDD from Seagate boasts 1 TB of storage, and 145,000 tracks per inch.

APPLICATION

Tracks are written on disks with Servo Track Writing air bearing spindles. A spindle holds a Hard Disk Drive (HDD) in a chuck adaptor, and can very accurately control the position and rotational speed of the HDD as the tracks are written onto them.

When writing tracks with spacing as small as 0.000003 inches (3 μ m), as most companies are today, there is nearly zero room for error. For work of this precision, the slightest vibration in the spindle could cause serious mistakes in track writing, which is why balancing of spindles used for servo track writing becomes so important.

If a spindle's vibration due to unbalance and other sources is greater than the track misregistration budget (the allowable inaccuracy), the HDD can experience error readings and bad track writing [6]. For precision spindles, a typical residual unbalance specification is 0.127 gram-meter at the front and back planes of the spindle.

In addition to the error from vibrations, unbalance can also cause noise, increased stress on the spindle, fatigue, shortened life, and more power use [7]. Unbalance is definitely worth taking the time and money to correct in spindles used for writing HDD tracks.

THEORY

Unbalance is caused by tolerances in the machining of parts, uneven finish on parts, nonhomogeneous density or porosity of parts, and asymmetry in design [7]. Some unbalance is unavoidable, and unbalance is statistically never the same for two parts [7].

Mathematically, the definition of unbalance is the product of the mass and the distance between the center of mass and the center of rotation (equation 1).

$$U_{PM} = m r$$

EQUATION 1. *U- Unbalance, m-mass, r-distance between center of rotation and center of mass*

Unfortunately, when the spindle spins, the centrifugal force (equation 2) pushes the center of mass out, and away from the center of rotation, continually increasing the unbalance. [8]

$$F_c = m r \omega^2$$

EQUATION 2. *F=centripetal force, ω =angular velocity*

The total axial deflection can also be calculated using Equation 3 [4].

$$m(r+x)\omega^2 = kx$$

EQUATION 3. *k=stiffness of the shaft, x=deflection of the shaft.*

METHOD

The spindle is balanced on two planes, the front and back end. The front is balanced by removing mass from setscrews in the chuck adaptor. Setscrews are also used to balance the back.

A sensor is attached to each plane to be balanced, sensor 1 on the front, and sensor 2 on the motor in the back (Figure 6). The spindle is mounted in a compliant foam support and balanced at-speed.

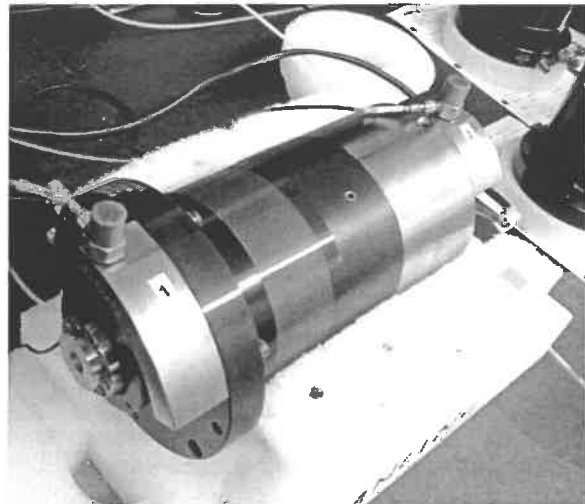


FIGURE 6. An air bearing spindle set up to be balanced.

The sensors are connected to a Nation Instruments data acquisition card. The software program calculates the angle from the top the heaviest part of the front and back are, and how much weight is offset at each (Figure 7).

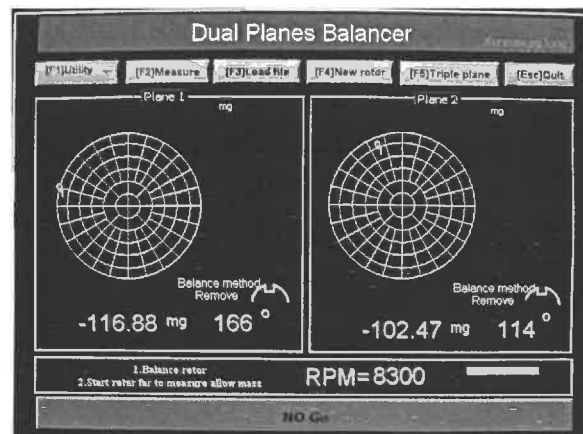


FIGURE 7. Operator's interface showing initial unbalance. The front needs 117 mg removed from 166° ccw, and the back needs 102 mg removed from 114° ccw.

Originally, there are six 145 mg setscrews in the front and six 190 mg setscrews in the rear. Of the six screws (all 60° degrees apart) the setscrews closest to the locations indicated are removed and either replaced with smaller ones or ground (Figure 8) to be slightly smaller in mass.

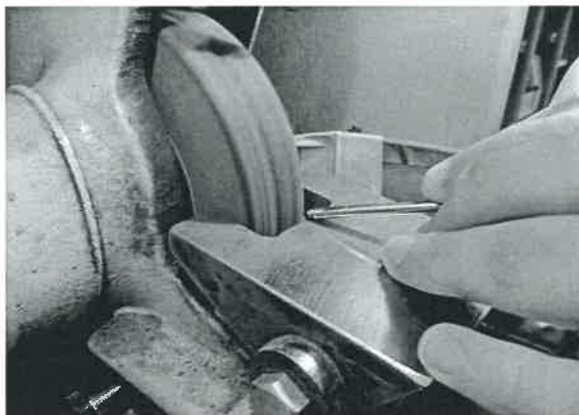


FIGURE 8. Buffing a setscrew removes a milligrams of weight during final balancing.

The spindles usually begin with unbalances of approximately 100-200 mg and setscrews are replaced with smaller ones until the unbalance is closer to 20-40 mg. At this point replacing them would make too much of a difference in weight so they are ground, taking off approximately 2 mg per swipe. When the unbalance is within 5 mg, only a buffer is needed to finish off the job, taking off mass smaller than that of a butterfly wing (approx. 3 mg) until the unbalance mass is within 2.5 mg (Figure 9).

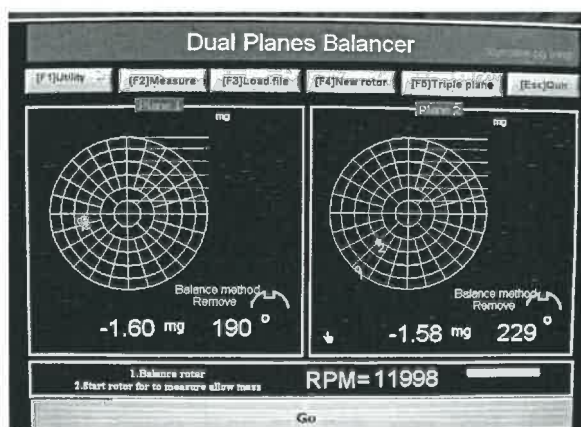


FIGURE 9. Computer screen of a balanced spindle (unbalance is less than 2.5 mg on front and back).

CONCLUSION

Spindles used for high precision applications such as servo track writing Hard Disk Drives must be balanced to achieve ever-increasing performance requirements associated with higher track densities. When writing upwards of 300,000 tracks per inch, the slightest vibration of the HDD can be a problem. To avoid this, the weights of setscrews in the front and back of each spindle are adjusted in order to obtain reduce unbalance.

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ASSEMBLING & TESTING A TAPERED ROLLER BEARING SPINDLE

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ABSTRACT

To achieve the full accuracy of tapered roller bearings, we measure alignment, compliance, and error motion during assembly. A roundness tester enables the alignment of reference diameters. An air bearing piston applies an axial load during rotation while an indicator measures displacement. Radial, axial, and face error motions are also measured, with a final correction to minimize twice-per-revolution error motion. For grinding, a 4RT tapered roller bearing spindle produces parts with form and finish approaching those made with air bearings but can carry much high loads.

MOTIVATION

The motivation for the 4RT came from a precision grinding application with a wheel force too high for air bearings to handle. The 4RT was designed with the same size, shape, and bolt patterns. Installation is easy because they require no compressed air system or oil pumps.

BACKGROUND

Tapered roller bearings have high stiffness and load capacity, making them an attractive alternative to air bearings—assuming they can be built into spindles with suitable accuracy [1]. The 4RT spindle is drop-in compatible with a 4R BLOCK-HEAD[®] and has found good use as a workhead for grinding applications. Conventional bearings can be used in precision applications to produce accurate results if they are set correctly in a spindle. At Professional Instruments, Timken cones and rollers are used with a custom-built one-piece, double cup stator.

ASSEMBLY

The stator has two concave conical surfaces called cups, and serves as the outer race for two Timken bearings. The rotor is a two piece shaft with heat-shrunk cone-roller assemblies. Twelve 10-32 socket head cap screws clamp the rotor halves together.



FIGURE 1. Roto-grinding on a 4RT spindle. 4RT spindles are suitable for precision grinding when forces are high and high speeds are not required.

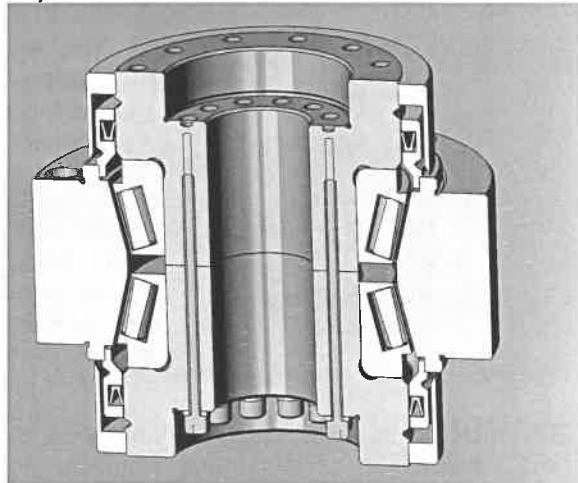


FIGURE 2. Cross section of a 4RT spindle.

The stators and rotor halves are sized with conical depth gauges in reference to a master part. Rotor halves are initially matched to a stator for a little extra end play. To assess the fit, stator-to-rotor axial displacement is measured while rotating with 200 pounds lifting the stator. (The rotor is bolted to the rotor of the test rig spindle.) For the optimum setting we look for linearity in the force versus displacement curves, shown in Figure 6. Experience has shown that



FIGURE 3. Measuring stator / rotor differences for assembly.

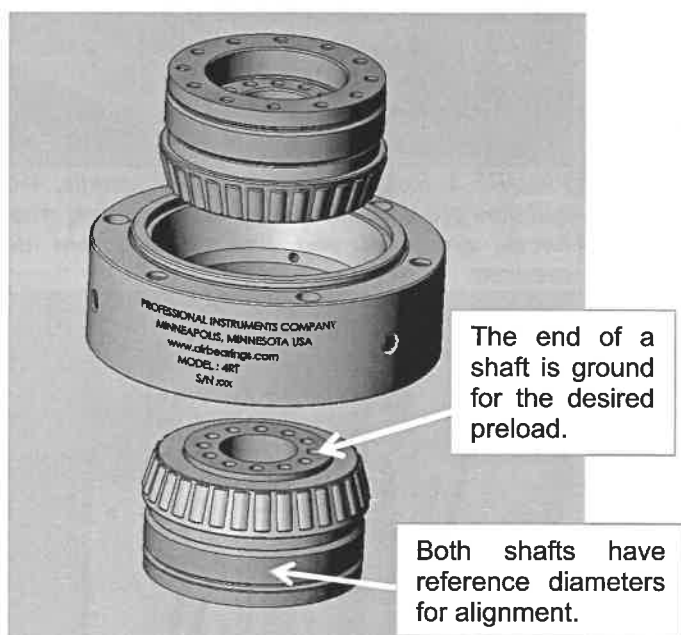


FIGURE 4. Stator and rotor halves of a 4RT.

200 microinches axial displacement gives the best combination of accuracy, stiffness, and running torque. Excess end play results in excess asynchronous error motion, especially at the cage rotation frequency of about 0.4 undulations per revolution. Adjustment is made by grinding the end of a rotor half. Fine adjustment is controlled by the tension on 12 clamping screws. Adjusting the force on selected screws minimizes the twice-per-revolution radial error motion. When both sets of bearings contact the stator, end play goes to zero. When the desired preload is reached, radial, axial, and face error motion is measured on the Figure 5 test rig.

End play becomes preload when all 48 rollers make line contact. The full load zone produces a stiffness of more than double that of a tapered roller bearing at 50% load zone. The line contact of tapered rollers increases the central stiffness up to six times higher than an angular contact ball bearing, and two times higher than cylindrical roller bearings [2]. While accuracy is also greatly improved, preload increases the friction and heat production, and limits speed.

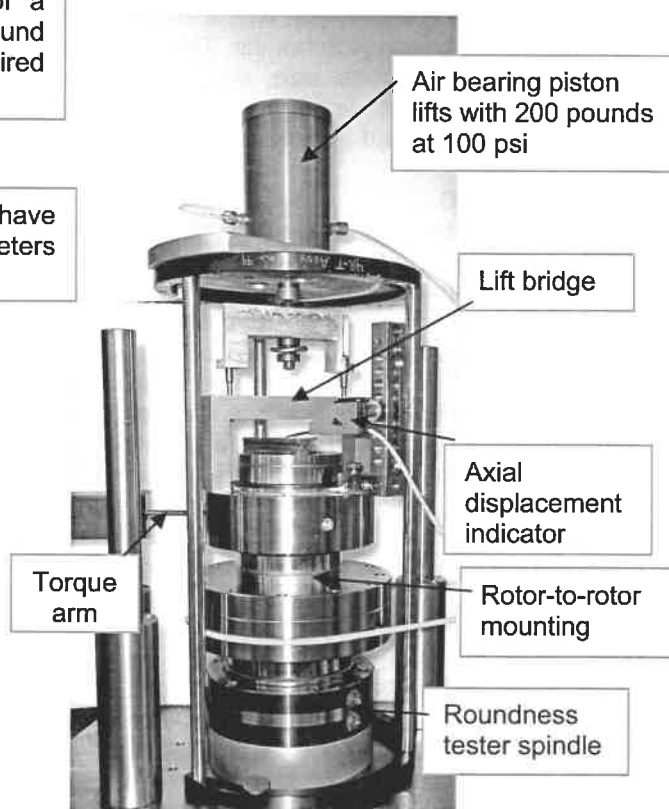


FIGURE 5. Test rig for alignment, compliance, and error motion.

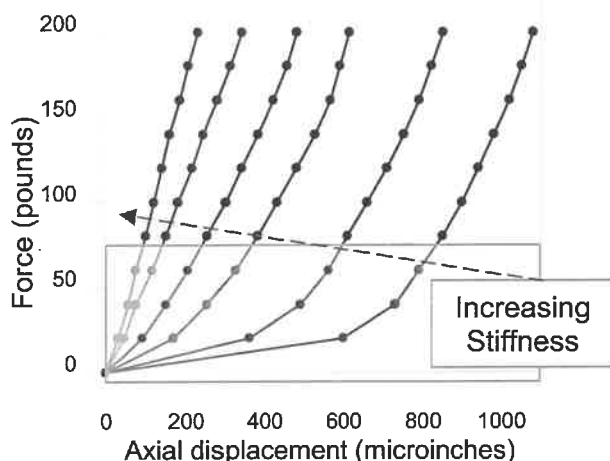


FIGURE 6. Plot of axial displacement as the shaft is ground shorter and clamped tighter. The improvement in stiffness is most apparent below 75 pounds.

Axial displacement is measured stator-to-rotor [3]. Stator rotation is prevented by a torque arm and a triangular gage block provides a target. The slopes in Figure 6 indicate bearing stiffness. The central stiffness or stiffness at the first point of loading, increases as the spindle is tightened. The design goal is to adjust the preload for maximum central stiffness without adding too much torque [3]. The screws are tightened in small increments to avoid kinking the shaft.

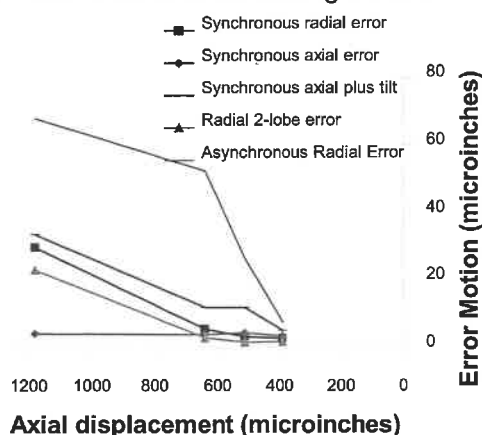


FIGURE 7. Effects of shaft grinding on end play and error motion. The ideal axial displacement for the grinding stage is 400 microinches with 20 pound-inches screw torque. Increasing to 40 pound-inches foreshortens the rotor for 200 microinches at 200 pounds.

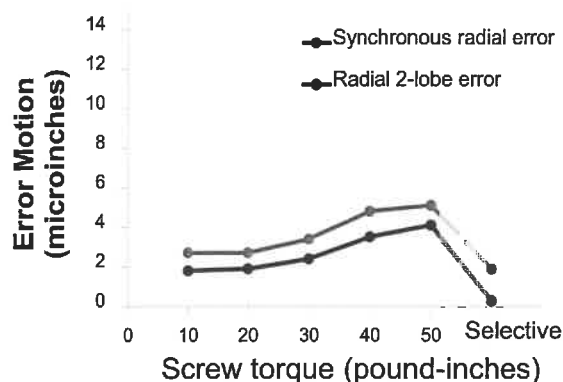


FIGURE 8. Final screw torque for certain screws is adjusted selectively to minimize two-lobe radial error motion.

TESTING

An assembled 4RT with the desired amount of preload is tested for radial, axial, and face error (axial plus tilt) motion [4]. The spindle rotor is bolted to a 4R air bearing rotor so that error motion can be measured with an indicator on the non-rotating stator of the 4RT. We ignore the error of the air bearing because it is under a microinch. A target plate is bolted to the stator for axial and face error motion tests as shown in Figure 9.

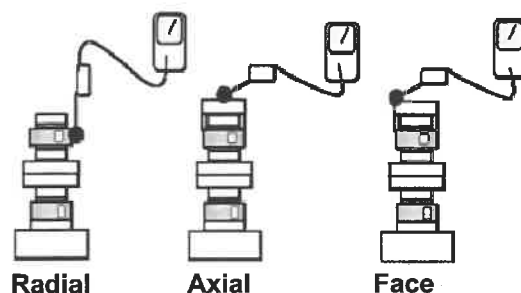


FIGURE 9. Locations for testing error motion.

TABLE 1. Typical results for the three error motion tests. Radial error is measured at the center of the stator. Face error is measured at a radius of 2.8 inches.

	Radial μ in	Axial μ in	Face μ in
Synchronous	2	2	5
Asynchronous	8	8	20
Total	10	10	25



FIGURE 10. Measuring radial error motion.

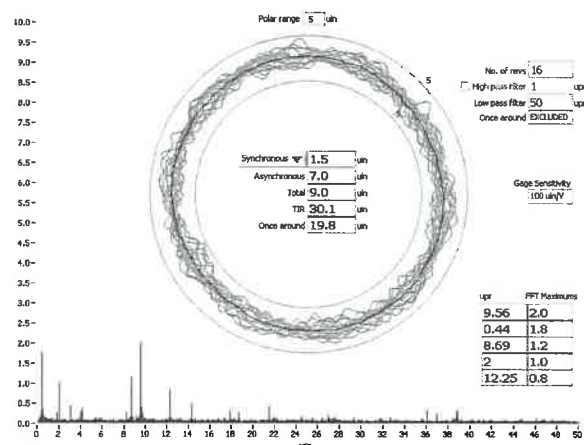


FIGURE 11. Representative results for a radial error motion test. In this case, synchronous is 1.5 μin and asynchronous is 7.0 μin after 16 revolutions.

Rolling-element bearings are subject to asynchronous errors, but when they are used in a grinder, the grinding process tends to average these out. The geometry of the workpiece is predicted by the synchronous error motion, not the total error motion. Every turn of the spindle cuts the high points that may have been missed before due to asynchronous error. This can also be seen in testing. As the revolutions included in a test increase, the true synchronous error is revealed with greater clarity. Single-point diamond turning is an event recorder, so asynchronous errors show in the surface finish.

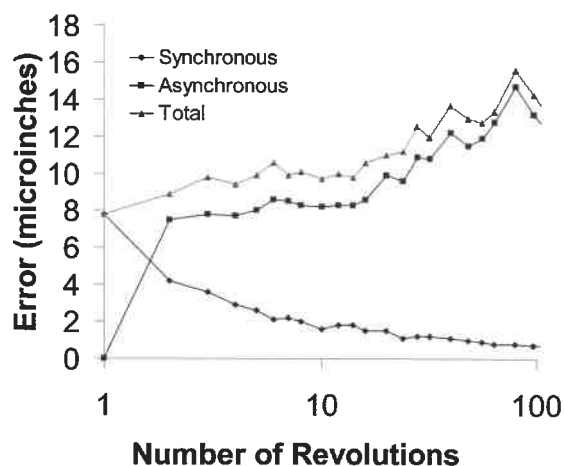


FIGURE 12. Synchronous motion is more clearly separated from asynchronous as the number of test revolutions increases [4].

CONCLUSION

To benefit from the accuracy built into the bearings, they must be assembled with correct alignment and preload. Optimum results are achieved by measuring preload and error motion as the spindle is assembled. Thus illustrating the toolmaker's adage:

If you can measure it, you can make it.

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AN AIR COMPRESSOR SYSTEM FOR PRECISION AIR BEARING SPINDLES

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ABSTRACT

Precision air bearing spindles due to their small air gap ($5\text{ }\mu\text{m}$ - $20\text{ }\mu\text{m}$) need to be supplied with a continuous supply of clean, dry and oil free air at a constant pressure. Compressing, cleaning and drying the air are important first steps to ensuring proper air quality. If measures are not taken to implement a correct air compressor system, condensation can cause rust on steel-based bearing surfaces as shown in Figure 1. There are many ways though to make an air compressor system more energy efficient and cost effective.

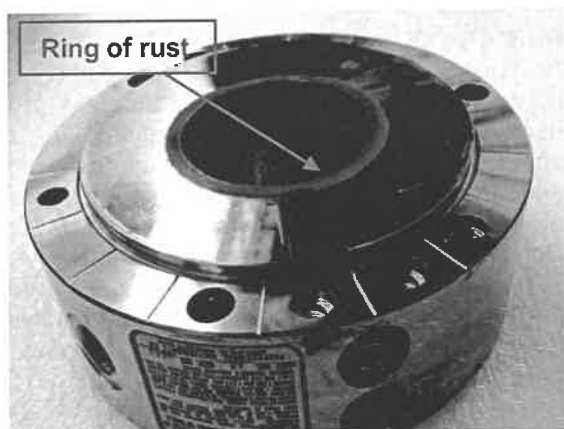


FIGURE 1. Rust due to condensation

COMPRESSOR

Multiple components make up an adequate air compressor system. There are two main types of air compressors; piston (single and multiple stage) and rotary screw. The rotary screw compressor in Figure 2 saves energy by optimizing the motor usage via variable speed drives [2]. Instead of always operating at full capacity, variable speed drives adjust motor speeds to meet the demand for compressed air [2]. This method increases the efficiency of a compressor by using the minimum power required at a given time [2].



FIGURE 2. Rotary screw air compressor with variable speed drives.

HEAT RECOVERY

Another way to save money and energy through a compressor is to use excess thermal energy to heat the factory. Directing the hot air either inside or outside can reduce the load on an air conditioner or supplement a heating system depending on location and season. Figure 3 displays a duct work system in sync with a baffle and thermostat that control where the heated air is released.

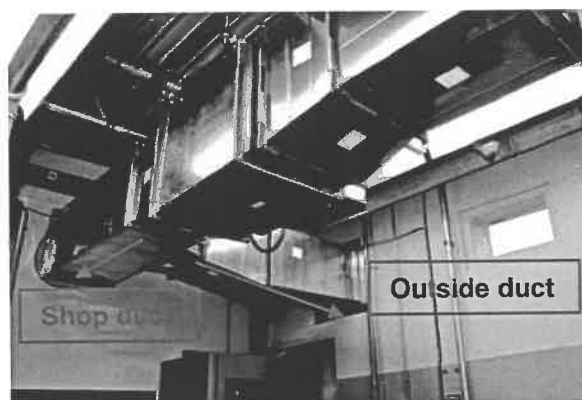


FIGURE 3. Ductwork directing the air flow either inside or outside.

Equation 1 calculates the annual energy savings from heat recovery.

$$AES = P \times hr \times L \times HRF \times C$$

EQUATION 1. AES means annual energy savings in \$, P is the motor's rated power in kilowatts, hr is the annual operating hours, L is the load factor, HRF is the heat recovery factor and C is the cost of electricity in \$/kwh [2].

Consider a compressor such as the one in Figure 2, rated at **P=45 kw**. Heat recovery from this compressor occurs only in colder months, about 6 months out of the year in this case. Consequently, an appropriate value for the annual operating hours would be **hr=4382.5** even though the compressor is always on. The unit does not always run at full capacity, therefore the average load is 70%, **L=0.7**. Assume **HRF=0.8**, meaning that 80% of the excess heat will be recovered [2]. Also assume the cost of electricity is **C=0.078 \$/kwh** [3]. For a compressor operating under these conditions, **\$8,614** could be recovered in thermal energy.

At Professional Instruments Creekside, we changed multiple parts of the building to decrease the amount of heat loss. A heat recovery system was installed into the air compressor output ducts, and a new roof and insulation was added to the building. Table 1 compares the Centerpoint Energy bill from 2003, before a heat saving system was installed, to the bill from 2010, after the modifications were made. Professional Instruments saved \$8792 from salvaging excess heat, and then keeping it within the walls of Creekside.

Table 1. Total money spent on gas before and after heat saving adjustments implemented [3]

Bill Date	\$	Bill Date	\$
1/30/2003	3333	1/4/2010	2304
2/28/2003	4214	2/2/2010	1782
3/31/2003	4088	3/5/2010	1030
4/29/2003	1770	4/2/2010	390
5/29/2003	539	5/3/2010	212
6/27/2003	60	6/2/2010	50
7/30/2003	49	7/6/2010	60
8/29/2003	48	8/2/2010	57
9/30/2003	48	9/2/2010	57
10/29/2003	177	10/4/2010	78
11/28/2003	764	11/2/2010	349
12/31/2003	1553	12/3/2010	1482
Total	16643	Total	7851
		Savings	\$8792

PRIMARY CLEANING AND DRYING

Air leaving the compressor contains particles of oil and water. In addition, small amounts of oil used in the compressor leak out into the air stream. Figure 4 displays the primary cleaning and drying system components. Cleaning the air begins in Unit A with a water separating filter. Unit B uses both activated carbon granules and a 1 µm coalescing filter [4]. Next the air enters the refrigerant dryer, Unit C. This dehumidifying device uses refrigerant in heat exchangers to cool the air condensing water from the air.

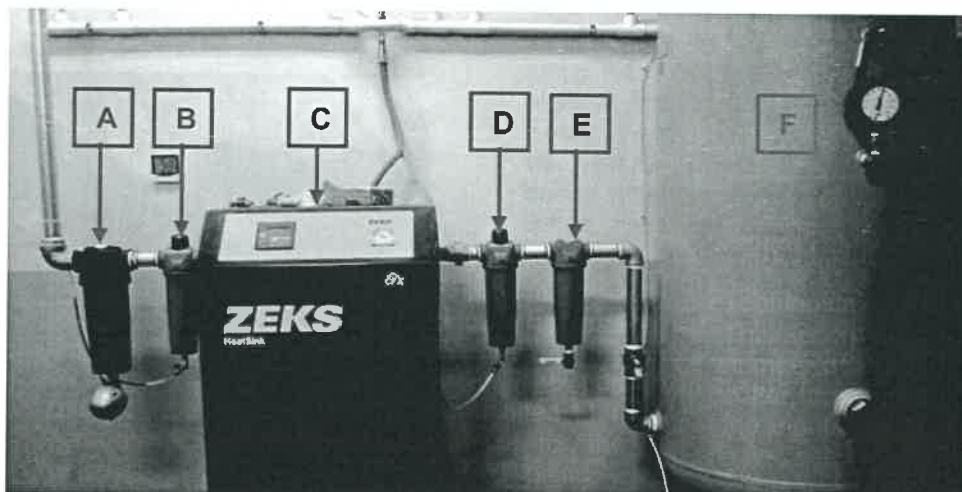


FIGURE 4. Oil filters, heat exchanger dryer, and storage tank

Another set of filters acts as a backup to catch any remaining particulates, oils, water and aerosols. Unit D contains a 0.01 μm coalescing filter and Unit E is the final filter in the primary cleaning system. The air then enters a storage tank, Unit F, also shown in Figure 4.

SECONDARY CLEANING AND DRYING

Air leaving the storage tank then travels through a secondary or backup cleaning and drying system. Figure 6 illustrates a second moisture removal component, oil filters and air flow controls into the shop ready to use. This dehumidifying device uses alkali chloride salt tablets, shown in Figure 5, that absorb water vapor and reduce the air dew point by approximately -7°C below the air inlet temperature at the desiccant tower.



Figure 5. Deliquescent desiccant tablets [5]

This method of water vapor removal requires no power making this part of the air compressor system beneficial for both the bearings and the check book [5].

Although the first dryer should do all of the work, the main purpose of a second dryer is to provide a reliable back up when the primary drying system fails. When maintenance or electrical shortages limit the desiccating powers of the primary dryer, the non-electrical dryer will provide air sufficiently dry enough to run precision air bearings temporarily. The secondary dryer is always on and fails at a gradual rate instead of instantly like the refrigeration unit. For high value machines and systems, having redundant drying equipment can save enormous amounts of time and money by preventing failures. As well, in these types of systems, incorporating a precautionary nitrogen tank can be useful. When an alarm sounds due

to a pressure or dew point change, a nitrogen tank can be used for a short period of time to run the air bearing. Figure 6 also includes a discarded water tank, which separates the oil from the water to meet drainage regulations.



FIGURE 6. dryer, oil remover, air controls, water tank

OTHER WAYS TO SAVE

There are more opportunities than those previously discussed to make an air compressing system more efficient. One of the largest sources of wasted energy in an air compressing system is in air leaks [2]. Executing regular leak checks and rapidly repairing holes is important for maintaining an efficient system. Decreasing the temperature of the inlet air into the compressor also increases the effectiveness of the compressor [2]. The cooler the air the less expansive it is, requiring less work from the compressor to compact the air [2]. At Professional Instruments Creekside we reduce the air pressure to 40 psi at night lessening the load on the air compressor. We also recently enlarged our air pipes, shown in Figure 7, from 12.70 mm diameter to 31.75 mm diameter to improve flow. Using properly sized piping increases the efficiency of an air compressor system by decreasing the pressure drop throughout the pipes from the regulators to the bearing.



Figure 7. 31.75 mm diameter piping

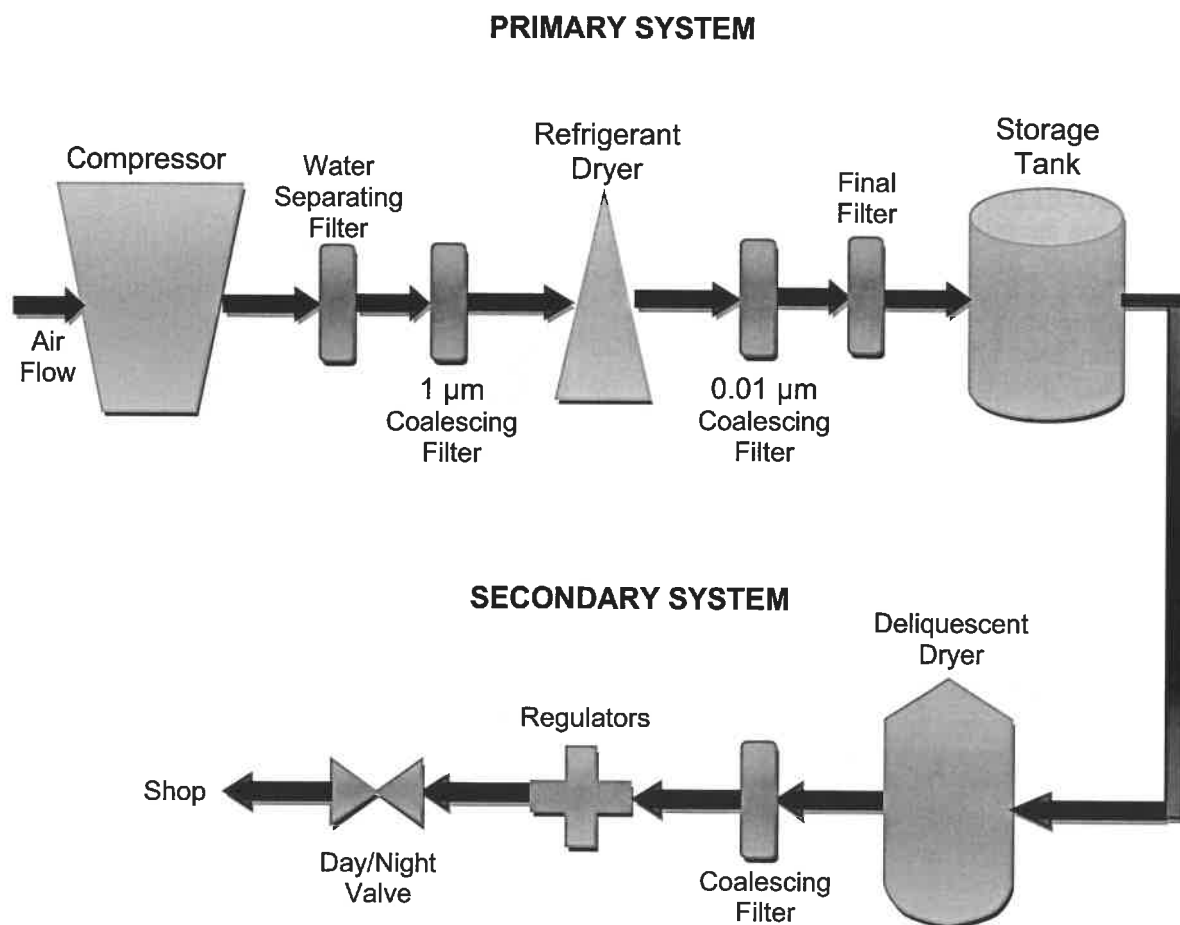


Figure 8. Flow chart of primary and secondary compressing, cleaning and drying systems

CONCLUSION

Figure 8 exemplifies Professional Instrument's system to ensure air free of oils and moisture, which is required to run smooth and long lasting precision air bearings. Although there are many different elements, each has a function that is vital to the performance of precision air bearings and must not be overlooked. No matter where compressed air is being used, water will always be present. Air bearings cannot operate properly with condensation; therefore the use of two air drying units is critical in preventing downtime [5]. Implementing proper mechanisms for compressing and transporting air will prove fiscally advantageous and add to the green initiative.

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A NEW MACHINING COST CALCULATOR (MC²)

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INTRODUCTION

A common objective in manufacturing research is to increase productivity and efficiency while decreasing cost. In machining, dramatic gains in productivity were realized with the introduction of computer numerically controlled (CNC) machines. Further advances in computer technology led to programs that could provide three-dimensional renderings of parts before they were machined. To complement these programs and to aid in the generation of the required code for CNC machine tools, computer aided manufacturing (CAM) software was developed. This software also supplies visualization of the machining steps and calculation of the time required to complete the CNC program. With this time estimation, users are able to select efficient strategies that reduced cost by reducing machining time. However, this time is calculated without considering the time required to accelerate to the desired feed rate and decelerate to a stop.

To provide a more accurate time estimate and, ultimately, improved cost prediction, a Matlab-based graphical user interface (GUI), entitled the Machining Cost Calculator (MC²), was developed that considers the acceleration and deceleration times in the calculations. The GUI consists of a user-selected internal pocket shapes and machining strategies.

TIME CALCULATIONS

Traditional time calculations in CAM software are completed by dividing the move distance by the user-selected feed rate. In the MC² time calculation there are multiple steps that are followed to respect the actual acceleration/deceleration values. First, it must be determined over what distance the acceleration will occur. Equations 1 and 2 are used to calculate this distance.

$$t = \frac{v_1 - v_0}{a} \quad (1)$$

$$d_1 = \frac{1}{2}at^2 + v_0t + d_0 \quad (2)$$

In Eq. 1 v_1 is the commanded feed rate, v_0 is the initial feed rate, a is the machine axis acceleration, and t is the time required to reach the commanded feed rate. In Eq. 2 d_1 is the distance to reach constant velocity and d_0 is the starting position. For all line segments within the part path, it is assumed that the starting velocity and position are zero.

Once d_1 is calculated, it is compared to the commanded distance. If d_1 is greater than half of the total move distance, then the tool does not reach the desired feed rate. In this case, the time to move the required distance is calculated using Eq. 3,

$$t = \frac{d}{a} \quad (3)$$

where d is half the commanded distance. However, if the distance required to accelerate to the desired feed rate is less than half the total move distance, then the commanded feed rate is reached for only a portion of the move. In this case, the distance over which the commanded feed rate occurs and the corresponding time must be calculated. To calculate the distance of the full feed rate portion of the move, twice the distance to accelerate to the feed rate is subtracted from the total move distance. The result is the distance over which the tool will travel at the commanded feed rate. Then, this distance is divided by the feed rate to calculate the associated travel time. With this value, the total move time is calculated by summing the full feed rate time with twice the time required to accelerate to the feed rate. The time to accelerate to the feed rate is doubled to account for the acceleration and deceleration time, where acceleration and deceleration times were assumed to be equal. Finally, once all of the times to complete the commanded pocket path are calculated, these times are summed to

determine the total time, t_m , required to machine the pocket given the user-selected machining conditions.

COST CALCULATION

The final output from the program uses the calculated machining time to compute the machining cost based on user inputs. Equation 4 describes the cost equation. The cost per part, C_p (\$), is calculated using the machining time, t_m (min), and user input values for the machining rate, r_m (\$/min), the tool change time t_{ch} (min), the cost per tool, C_t (\$), and the tool life, T (min), for the selected cutting conditions.

$$C_p = t_m r_m + \frac{(t_{ch} r_m + C_t) t_m}{T} \quad (4)$$

Table 1. Calculated and actual times for the test pocket when including acceleration.

1000 mm/min			
Strategy	MC ² time (sec)	Actual time (sec)	% difference
Straight line	453	452	0.22
Zig-zag	325	327	0.61
Spiral-in	231	233	0.86
3000 mm/min			
Straight line	195	210	7.14
Zig-zag	118	128	7.81
Spiral-in	84	89	5.62

VALIDATION

To validate the program a set of tests were completed to compare the calculated machining time (with and without considering acceleration) to the actual time required to machine a pocket on a Mikron UCP-600 Vario CNC machining center. The selected internal pocket 50 mm square and 10 mm deep. The tool was a 10 mm, four-flute square endmill. The spindle speed was 5000 rpm and two feed per tooth values of 0.1 mm/tooth and 0.3 mm/tooth were applied. The rapid plane height was specified at 10 mm above the part surface and the rapid velocity was 0.33 m/s. Average acceleration values of 1.08 m/s² and 1.53 m/s² were measured independently using an accelerometer and used for the two feed rates of 1000 mm/min and 3000 mm/min, respectively. Tables 1 and 2 display the results and provide a percent difference to show the improvement in the accuracy of the time estimation using the acceleration-based approach.

Table 2. Calculated and actual times for the test pocket when acceleration is not considered.

1000 mm/min			
Strategy	Infinite acceleration time (sec)	Actual time (sec)	% difference
Straight line	393	452	13.1
Zig-zag	317	327	3.07
Spiral-in	225	233	3.43
3000 mm/min			
Straight line	143	210	37.9
Zig-zag	106	128	17.2
Spiral-in	75	89	15.7

Based on these results, a clear improvement is seen when including acceleration in the machining time calculation. By more accurately calculating this time, a more accurate cost estimate can be made enabling manufacturers to better predict part costs.

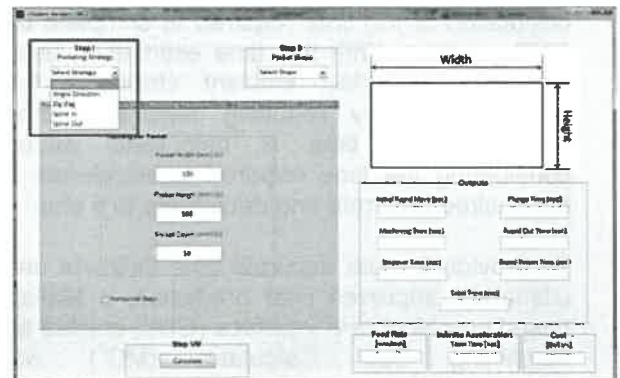


Figure 1. Pocketing strategy selection.

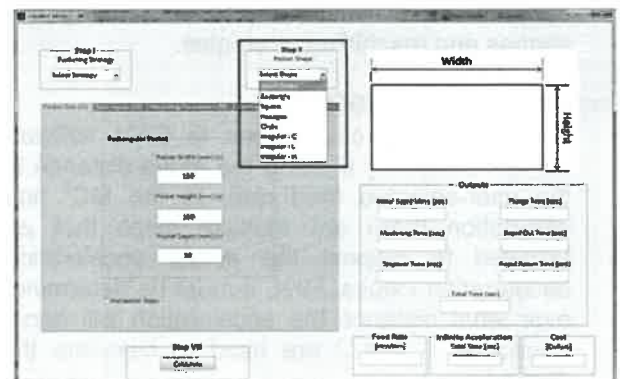


Figure 2. Pocket shape selection.

MC² DEMONSTRATION

In this section, examples are provided to demonstrate the MC² GUI interface, including the inputs and outputs. MC² enables the user to

input the required information and select the pocket geometry and strategy. Based on the user input, the machining time and cost is estimated. The first input, labeled Step I, appears as a drop down menu. This menu enables the user to select from the four machining strategies, shown in the red box in Fig. 1 (straight line, zig-zag, spiral-in, and spiral-out). The second drop down menu allows one of seven pocket shapes to be selected (rectangle, square, circle, hexagon, H, C, and L). The menu is labeled Step II and is identified by the box in Fig. 2.

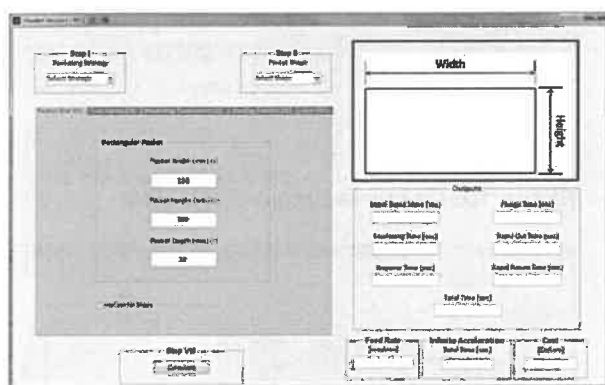


Figure 3. Pocket shape graphic.

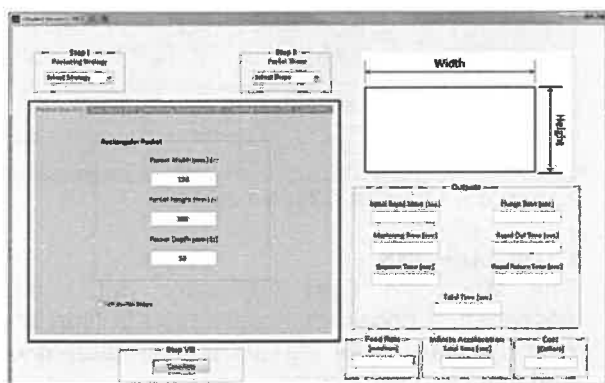


Figure 4. Pocket size inputs.

Based on the selected pocket shape, the upper right figure switches the image to show the dimensions required to fully define the pocket. Figure 3 identifies this image. The third user input is the first tab on the dark grey panel and it changes its display to enable the proper inputs for the pocket shape depicted in Fig. 3. Figure 4 displays the tab, labeled Pocket Size (III), for the rectangular pocket in this example. The part path for the selected pocket is exhibited in a separate figure window. An example for the rectangular pocket to be machined using a spiral-in strategy is provided in Fig. 5.

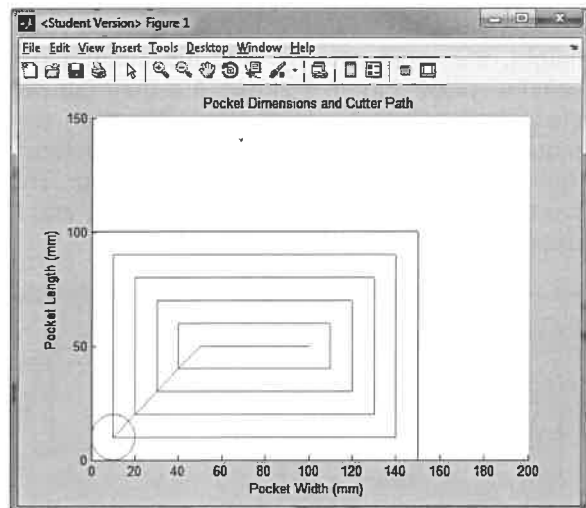


Figure 5. Path for a rectangular pocket with a spiral-in strategy.

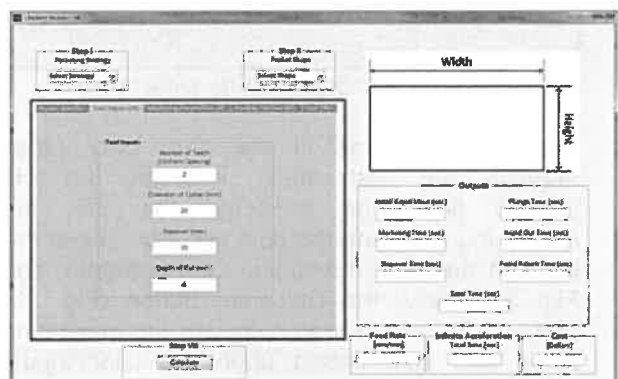


Figure 6. Tool definition in Step IV.

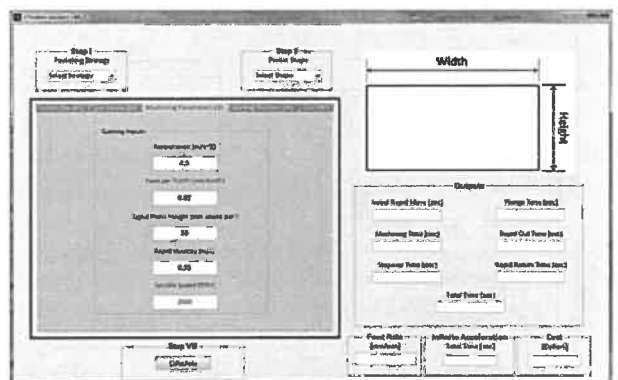


Figure 7. Machining parameters for Step V.

Tool Inputs (IV) is the fourth step in the program. This second tab requires the user to define the number of teeth on the tool, the diameter of the cutter, the stepover, and (axial) depth of cut. All the inputs are labeled with their accompanying units; see Fig. 6. The Machining Parameters (V) tab is used to input additional specifications for the operation, including the machine axis

acceleration, the feed per tooth, the rapid plane height, the rapid velocity, and the commanded spindle speed. Figure 7 shows this third tab with the default values for each input. The sixth step requests the location initial tool location before it approaches the part to begin machining. This fourth tab is labeled Starting Position (VI) and is displayed in Fig. 8.

Figure 8. Starting position entry (Step VI).

The right-most tab in the dark gray panel requests the cost inputs, including the tool change time, the expected tool life, the machining rate, and the cost per tool. These are used in the total machining cost estimate; see Fig. 9. Finally, the Calculate button (Fig. 10) prompts the program to calculate the machining times and cost based upon the user inputs identified in the previous steps.

Figure 9. Cost inputs (Step VII).

The outputs for the selected conditions are displayed in the lower right panel. The output times include the cutting and non-cutting times along with the total time of the operation. The output times depend on the selected machining strategy. Three other outputs are also displayed. The first is the feed rate, which is calculated from the user defined feed per tooth, spindle

speed, and number of teeth on the cutting tool. The second is the infinite acceleration time, or the time it would take to machine the pocket if the acceleration and deceleration times were neglected. Finally, the cost for the selected pocketing operation is displayed; see Fig. 11.

Figure 10. Calculate button (Step VIII).

Figure 11. Outputs supplied by MC².

CONCLUSIONS

The Machining Cost Calculator (MC²) was described. It considers acceleration in time and cost calculations for various pocket geometries and machining strategies.

ACKNOWLEDGEMENTS

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Design of a Cryogenic Target for Indirect Drive Ignition Experiments on NIF

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The National Ignition Campaign (NIC) began ignition experiments on NIF in FY2010. The culmination of target fabrications' efforts will produce an ignition target which meets or exceeds the physics requirements. The ignition target includes both a cryogenic base, which supports the electrical, thermal and gas fill connections to the ignition target inserter cryostat (ITIC), and the ignition hohlraum assembly which is supported by the Thermal-Mechanical Package (TMP). The target base contains a reservoir assembly which will carry the fuel and support filling the capsule in order to produce an ice layer. The ignition hohlraum design contains windows for viewing the ice layer with x-ray imaging systems. We will discuss ignition target requirements and how these have been met with current target design, fabrication, assembly and metrology.

LLNL-ABS-420716

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MISALIGNED BEARINGS CAUSE OVAL ERRORS AND SIZE VARIATIONS

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INTRODUCTION

Since nothing is perfect, all spindles have some amount of twice-per-revolution radial error motion in which the axis of rotation travels in an oval orbit around its axis average line. The result is two Undulations Per Revolution (UPR) out-of-roundness for lathe cuts and fixed sensitive direction measurements. A related effect is variation in once-per-revolution displacements for rotating sensitive direction applications. The root cause could be many things; here we discuss the case of bearing misalignment due to bending a rotor shaft.

MOTIVATION

Since the problem is ubiquitous it's worth a closer look. This specific error source has the interesting consequence of size variation rather than variation of location.

EXAMPLES

Misaligned tapered roller bearings. Tightening a bearing locknut. Consequences for turning and for boring.



FIGURE 1. Before and after aligning bearings: Misalignment creates two rotating pinch points. Two-lobe radial error motion is one result. Improving the alignment reduces the two lobe.



FIGURE 2. Bearing locknuts induce off-axis strain: Tightening the nut inevitably bends the shaft, despite heroic efforts to square up the shoulder to the threads.

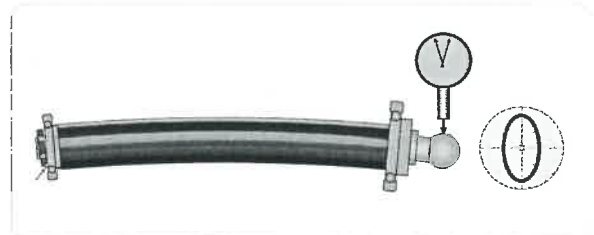


Figure 3. A twice-per-revolution radial error motion results from bearing misalignment. We will assume that this is the only error source as we contemplate some of the consequences.

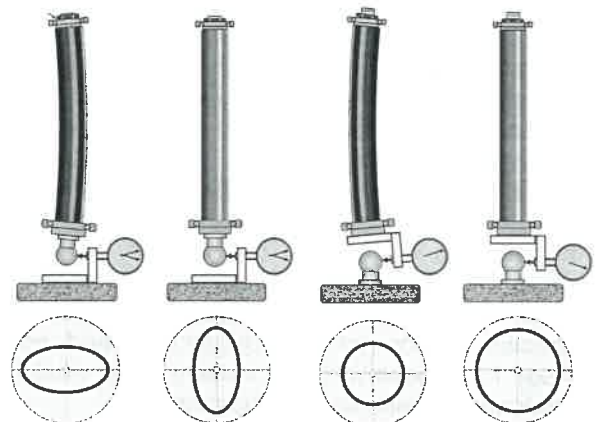


Figure 4. Four orientations:

- Fixed sensitive direction with the bend in line with the indicator.
- Fixed sensitive direction, bend 90° to indicator.
- Rotating sensitive direction with the indicator in line with the bend.
- Rotating sensitive direction with the indicator 90° to bend.

With the indicator looking in at the bent rotor shaft we see two highs and two lows as the pinch points rotate. With the indicator rotating with the shaft we see a constant value. Indexing the indicator 90° to the bend also gives a constant-but-higher value since the distance from the axis average line is less. Since this hypothetical stator is perfect, the rotating sensitive direction polar plots are perfect circles.



Figure 5. The two-indicator method gives us another view: X and Y indicators show twice-per-revolution displacements 90° apart, revealing an oval orbit [1].

Combining X and Y gives a once-per-revolution sine wave with the same amplitude as the oval. This does not represent a centering error; it is a measure of the amount of two-lobe and a signal that the two-lobe drops out when the sensitive direction rotates. The twice-per-revolution motion is also contained in the rotating sensitive direction but now it exhibits as once-per-revolution content which changes with the orientation angle relative to the bend in the rotor shaft. In other words, a two-lobe error is seen one way when the reference coordinate frame system turns with the rotor and a totally different way when it is fixed to the stator. Assuming perfect circular elasticity, this spindle will bore round holes. The bend in the rotor shaft goes unseen except for the hard-to-find effect of the change in the size of the orbit of the tool bit as it is (theoretically) perfectly indexed. In this thought experiment we invoke a boring head with ability to index without radial error so that the size change is all due to the bent rotor. The consequences of a bent rotor shaft are an oval orbit, two-lobe radial error motion from the outside looking in at the rotor, no radial error motion when viewed from the rotor looking out at the stator, holes bored round and on-location but different diameters, and variation in once-around as seen with X and Y indicators.

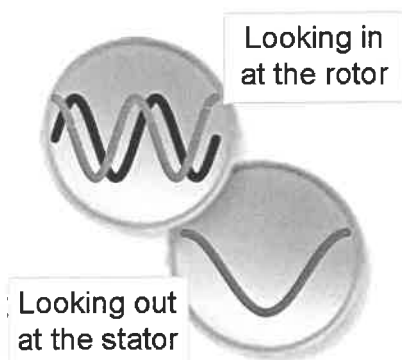


Figure 6. Two sides of the same coin.

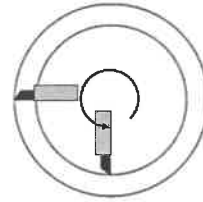


Figure 7. A boring bar with one bit in-line and one 90° to the bend—assuming they are set to the same radius—will bore two different diameters but both will be on the axis average line.

Once-around sine waves are typically due to eccentricity of an artifact but here the signal represents the difference between the major and minor axes of the orbit of the axis of rotation. In other words, the bent rotor shaft does not affect roundness or location.

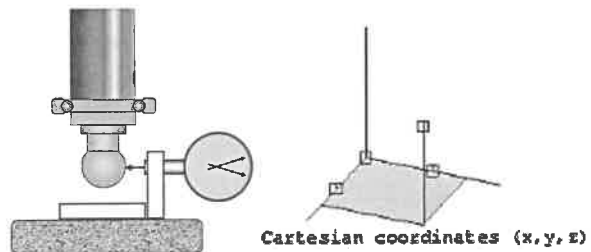


Figure 8. Coordinate system for fixed sensitive direction. Imperfections of the rotating structures are seen as they pass the point of observation.

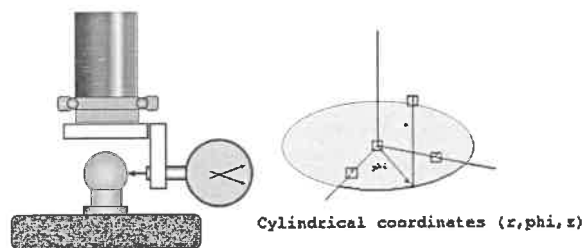


Figure 9. Coordinate system for rotating sensitive direction. The view is from a point on the rotor looking out at imperfections of stationary structures.

An error in one coordinate system can go unseen in another. The error does not evaporate, it switches to a non-sensitive direction. X and Y indicators detect all in-plane radial motions but the choice of which display to use determines the shape of the plot.

CONCLUSIONS

The presence of twice-per-revolution motion tells us to expect a like amount of variation in the once-around displacement since they are two sides of the same coin. A usual byproduct of a oval orbit is seen as a centering variation but in this case it is size variation. Real-world axes have more than just one error source. Your mileage will vary.

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THICKNESS MEASUREMENTS OF THE GAS DISTRIBUTION LAYER FOR A DIRECT METHANOL FUEL CELL

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INTRODUCTION

The subject of an ongoing collaboration between the University of North Florida (UNF) and the University of Florida (UF) is the development of a new direct methanol fuel cell. The objective of the research is to create a cell that can power portable electronic products with increased energy density and portability.

In current commercially-available direct methanol fuel cells, water filtration from the cathode to the anode side of the cell is performed by an active process. This process requires large, power-consuming components such as a water pump, water separator, and condenser. UNF has developed a gas distribution layer (GDL) which performs the water filtration tasks passively. This passive water filtration enables the elimination of several components found in a conventional fuel cell, which, in turn, increases the volumetric energy density to levels comparable to lithium ion batteries.

To support this development effort, UF has performed measurements on the GDL to determine the manufacturing quality. The GDL is comprised of two components. The first is carbon fiber paper (CFP) that controls the amount of oxygen flowing through the system. The second is an ink coating that is applied to the CFP and provides the necessary water filtration in the cell. The system performance is sensitive to the thickness of the GDL. If it is too thick, then too little oxygen passes through. Conversely, a thin GDL allows more water than desired to flow through the system.

This paper describes the design and calibration of the thickness measurement device and example measurement results. The thickness was determined using a capacitance probe

setup which measures the voltage differential between a reference point and the sample. The GDL was held in place by a vacuum chuck that was designed and built at UF to hold the samples flat during testing. Thickness maps of GDLs (120 mm × 220 mm) are provided in order to characterize the GDL thickness variation. A variation in the thickness as a function of ink loading was observed. Additionally, a validation of the capacitance probe setup was conducted using a micrometer.

EXPERIMENTAL SETUP

Capacitance probe

Capacitance probes enable non-contact measurements, a requirement for the thickness evaluation of the GDL. These sensors measure the difference in capacitance between a reference and sample by applying an alternating voltage to each conductive object and measuring the response. The response is inversely proportional to the distance between the sensor and target. The change in capacitance is multiplied by an appropriate calibration factor to determine the thickness of the sample.

Vacuum chuck

It is necessary that the target surface be as flat as possible during measurement to avoid convolving this non-flatness with the actual thickness variation in the GDL. To achieve this, a vacuum chuck was developed which enabled the GDLs to be uniformly suctioned against its surface. To cover the 120 mm × 220 mm GDL surface area, an aluminum block was machined with over ten thousand 0.8 mm diameter holes; see Fig. 1. The small hole diameter was chosen because the GDL was flexible enough that any larger-sized hole would have caused localized errors in the thickness measurements. As seen in Fig. 1, the chuck was placed on two stacked

programmable stages to enable capacitance difference calculations to be completed.

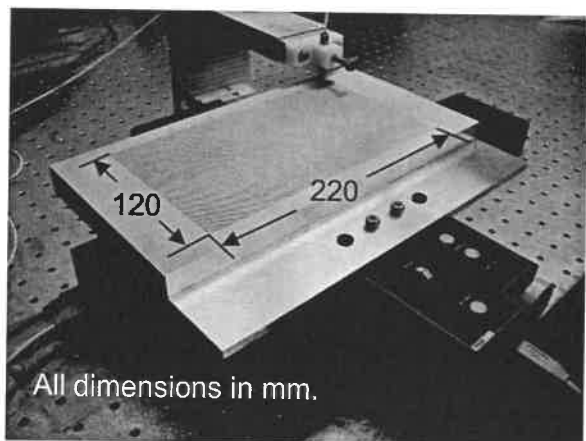


FIGURE 1. Vacuum chuck located on two stacked positioning stages. The capacitance probe is also shown.

RESULTS

To determine the GDL thickness maps (see Fig. 2), two measurements were performed. First, the vacuum chuck flatness was determined by scanning the measurement area. Second, the GDL was inserted and measured. The two measurements were then differenced to isolate the GDL thickness. The capacitance probe sensor used for this project had a 3.2 mm sensing diameter over which it averaged the capacitance of the sample. Considering this averaging effect, the stage was stepped along the y-axis in 1.5 mm increments and scanning was initiated in the x-axis at a rate of 37,000 samples per line (220 mm distance).

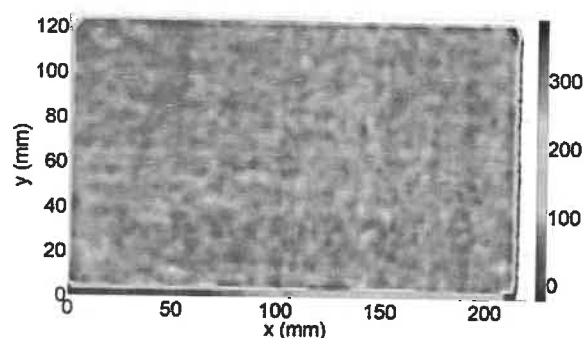


FIGURE 2. Surface map of an eight ink coat GDL. Colorbar units are in μm . The GDL had an average thickness of $282.7 \mu\text{m}$ and a standard deviation of $13.2 \mu\text{m}$.

Another GDL feature of interest was the thickness change with progressive ink loading. Eight individual ink coats were applied to the CFP and the thickness between each subsequent coat was measured. See Fig. 3. In the ink coating process, the carbon fiber paper is cut from a large roll down the middle to create both a left and a right piece. A systematic difference in barrier thickness between the left and right piece was observed.

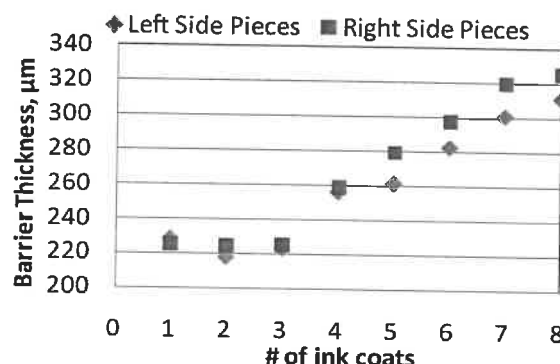


FIGURE 3. GDL thickness versus number of ink layers.

The constant variation was investigated by conducting performance tests on GDLs from each side of the roll. These tests are comprised of a diffusion test, which measures the methanol diffusivity of the cell, and a capillary pressure test that quantifies the water transport capacity. The capillary benchmark is the easier of the two to satisfy and each side of the roll passed every time. The methanol diffusion is the more difficult target to achieve and could fail despite having conducted the painting process in the correct manner. It was found that simply adding a thicker coat or painting nine coats of ink on the left side pieces would overcome this challenge. A total of 22 GDLs were scanned and the mean thickness and standard deviation for each are presented in Table 1. GDLs 1-16 were measured as part of the progressive ink loading experiment, while GDLs 17-22 had the full eight ink layers.

TABLE 1. GDL thickness data.

GDL #	Mean Thickness (μm)	Std. Deviation (μm)
1	228.6	12.9
2	225.1	12
3	218	12.1

4	224.5	11.4
5	223.3	13.2
6	225.1	12.2
7	256.2	13.8
8	258.9	14.3
9	261.5	13.5
10	279.2	13.5
11	282.7	13.2
12	297.7	13.1
13	311.5	14.9
14	325.6	14.3
15	301	13.7
16	319.9	13.6
17	286.5	14.8
18	276.9	13.8
19	301.7	14.1
20	245.4	12.7
21	279.7	12
22	277.9	12.1
Average		13.2

COMPARISON TESTS

The measurement technique presented is unique due to the vacuum chuck flattening. Therefore, a validation of the capacitance probe data was needed to confirm the results. It was found that the GDL could not be directly compared to scanning white light interferometer (SWLI) results due to the lower lateral resolution of the capacitance probe data. An example of the GDL SWLI data is given in Fig. 4.

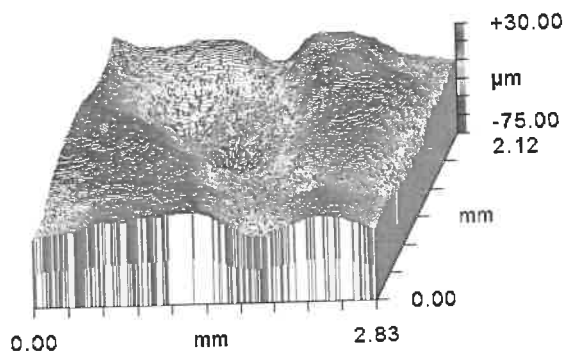


FIGURE 4. SWLI height map of GDL. The scanned area is 2.83 mm long by 2.12 mm wide.

As seen in Fig. 5, the capacitance probe data does not have comparable resolution to the

SWLI measurement. The data displays a smoothing effect due to the averaging over the capacitance probe sensing area.

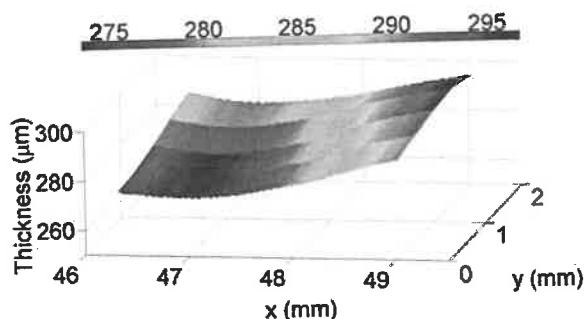


FIGURE 5. 3-D plot from the capacitance probe data. Colorbar units are in μm . This image is a small fraction of the size of Fig. 2.

Validation was conducted by measuring a sample of known thickness. A 1.27 mm thick gage block was measured by three devices, including the capacitance probe setup (Fig. 6), a SWLI (Fig. 7), and a micrometer (Table 2).

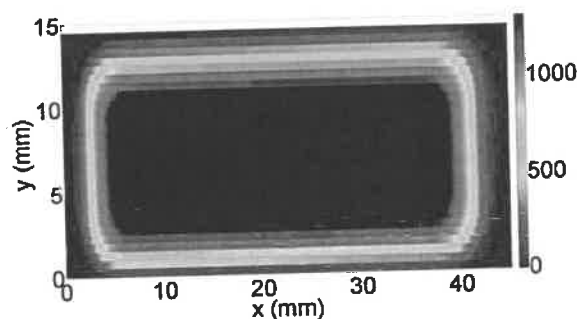


FIGURE 6. Height map of the 1.27 mm thick gage block and surrounding area from the capacitance probe setup. Note the averaging effect near the gage block edges

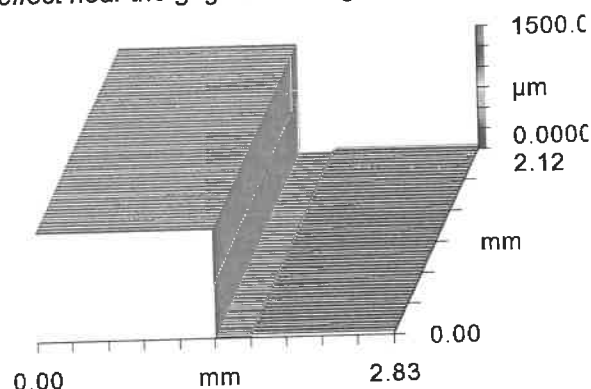


FIGURE 7. The 1.27 mm gage block thickness profile from the SWLI.

The mean thickness and standard deviation for each measurement type is included in Table 2. The micrometer result, which was averaged from three measurements at various locations on the block, was closest to the specified thickness of 1270 μm . The SWLI data was several micrometers thicker and was found by placing the 1.27 mm gage block on top of a second gage block and completing a 1500 μm upward scan. The capacitance probe results were 20 μm thicker than the expected value. All measurements were performed in a temperature controlled environment.

TABLE 2. Micrometer, SWLI, and capacitance probe thickness data of 1.27 mm thick gage block.

Measure Type	Mean Thickness (μm)	Std. Dev. (μm)	Percent Diff. (%)
Micrometer	1270	0.5	0
SWLI	1278.0	0.17	0.63
Cap. Probe	1290.2	22.5	1.59

One potential explanation for the SWLI measurement difference is that the gage blocks were not wrung together and debris was likely present between the two surfaces. This would lead to an increased thickness. For the capacitance probe data, it is believed that the calibration factor applied to convert the voltage into displacement was too large. This offset could be corrected for subsequent measurements. Also, the thickness was determined by differencing the voltage response of the gage block and the vacuum chuck surface. Small burrs or dust on the chuck surface would cause a larger gap between the sample and reference planes and lead to a thicker measurement. Polishing of the chuck surface would likely result in a more accurate measurement for the rigid sample. Note that this measurement was different than the GDL measurements because the GDL conformed to the chuck surface when the vacuum was applied.

DISCUSSION

A measurement apparatus was constructed to determine the thickness of the gas distribution layer in a direct methanol fuel cell. The device was developed to determine the manufacturing

quality and repeatability of the ink application to the surface of the GDL. It was found that the ink application is evenly applied to a standard deviation of 13.2 μm over the surface of the GDLs.

Alternate deposition methods, which include spraying and rod coating, will be investigated for two reasons. The first is to decrease the standard deviation of the thickness, thereby increasing the likelihood that the GDL will pass the diffusion and capillary performance tests. The second is a cost-benefit issue. The labor cost of the ink painting process is extremely high. GDL batches are currently manufactured in lots of eleven pieces. From these eleven pieces, four laptop-sized batteries can be constructed if each piece passes the performance tests. The formulation of the ink and deposition onto the carbon fiber paper takes, at minimum, 7 hours for one worker to complete. Design scale-up is a significant concern for this project and the painting procedure is ineffective for high volume production.

CONCLUSIONS

It was found that painting the ink onto the GDL yielded an average 13.2 μm standard deviation in thickness over the 22 tested samples. The painting technique is a satisfactory method for small scale production of direct methanol fuel cells. However, scale-up will require research into alternate ink deposition procedures.

ACKNOWLEDGEMENTS

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FREEFORM MIRROR FABRICATION BY MEANS OF FAST TOOL SERVO DIAMOND TURNING

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INTRODUCTION

Optical instrumentation for multi- and hyperspectral remote sensing such as telescopes or spectrometers are often based on metallic mirrors with reflective coatings for the desired application. In most cases where the applications wavelength spectrum ranges in the infrared, metal mirrors based on aluminum Al6061 substrates are diamond machined to fulfill the specifications regarding figure and roughness [1, 2].

Gebhardt et al. summarizes the fabrication chain for an all aluminum IR-spectrometer optics using the classical approach based on diamond turning of mirrors and gratings, the corrective machining, coating and the final system alignment under interferometric monitoring [3]. As in many other telescopes, the mirrors are rotationally symmetric to the optical axis of the instrument, although the surface section fabricated might be off-axis. Hence for design symmetry and manufacturing reasons the mirrors are diamond turned in their off-axis position [4].

Sweeney analyses different aspects of off-axis machining such as centrifugal forces, mounting forces and aggressive light weighting on surface figure and surface finish. He concludes in a general performance envelope among fabrication techniques that larger off-axis distances and more aggressive light weighting are leading to a higher figure deviation of the diamond turned surface. The surface roughness does not necessarily scale with the off-axis distance and degree of light weighting [5]. The minority of these shape errors can be corrected with the classical diamond turning approach, because the shape error correction is based on a modified 2D cutting profile of the diamond tool. Hence errors that are not rotationally symmetric to the turning axis can only be corrected with further polishing and figuring techniques that account for the error map as a surface information. But these processes are only target-aimed if the substrates are coated with an amorphous polishing layer such as nickel-phosphorous (NiP) prior to the diamond turning

[1, 6] or an amorphous silicon layer prior to the polishing, which might be a cost driver in telescope manufacturing.

This study is focused on a method to machine an off-axis mirrors with a high degree of light weighting in the center of the turning axis (on-axis). The expected benefits are:

- reduction of the surface figure deviation coming from centrifugal forces and temperature changes
- correction of non-rotationally symmetric errors

Challenges regarding the freeform machining, the referencing for the measurement as well as tilted mechanical interfaces for the mounting are discussed and solutions for a hybrid manufacturing consisting of diamond turning and diamond milling are presented.

OFF-AXIS - FREEFORM MIRROR TRANSFORMATION

The freeform fabrication of metal mirrors for imaging applications is increasingly discussed by optical community. The technical advancements allow an accurate freeform manufacturing process [7, 8, 9]. Another enabler is the implementation of freeform surfaces into optical design tools.

Freeform surfaces are build into systems to reduce the number of elements or to shift the position of mirrors to save space and weight. Freeform mirrors are also used to reduce aberrations or to achieve a required optical function such as an anamorphic imaging. Moreover classical off-axis mirrors can be handled as freeform surfaces regarding the manufacturing process, the metrology and the data handling. The necessary transformations are based on the two steps shown in figure 1. First the center of the mirror body is shifted onto the axis of rotation (fig. 1 (b)). The value for this translation is often given by the off-axis distance in the optical design tools. The second transformation is a tilt about an axis perpendicular to the axis of rotation to minimize the

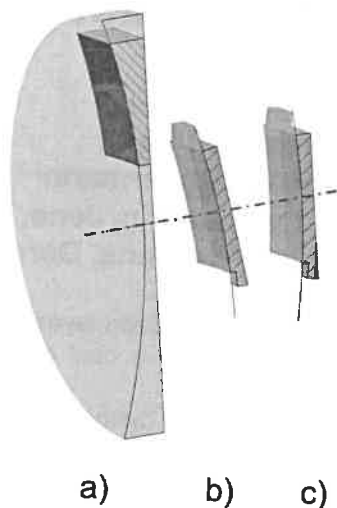


FIGURE 1. On-axis freeform geometry is derived from two transformations of the off-axis mirror.

slope needed to machine the mirror (fig. 1 (c)). Different optimization approaches can be used to calculate the angle. A proven method is to numerically minimize the root mean square (rms) deviation of the freeform surface from a best fit asphere, which is rotationally symmetric to the axis. The example discussed for the fabrication of an off-axis mirror as a freeform mirror is derived from an IR-Telescope design of an unobscured afocal folded Three Mirror Anastigmat (TMA) optical design as described in detail in a previous publication [4]. Here the primary mirror has a Clear Aperture (CA) of $100 \text{ mm} \times 101.7 \text{ mm}$ and an off-axis distance of 116.2 mm . The aspherical surface is shifted and tilted about 10.4° to be diamond turned on-axis. The freeform share is shown in figure 2. The deviation from the rotationally symmetric shape is $220 \text{ }\mu\text{m}$ peak to valley (p-v). This surface point cloud is used to calculate the radius compensation of the diamond tool and to generate the cutting location data for the freeform turning process.

REFERENCES FOR MEASUREMENT AND SYSTEM INTEGRATION

For the measurement of the surface and for the final system assembly further reference marks and interfaces with a precise relation to the freeform surface are mandatory to detect and align the coordinate system of the mirror surface. They have to bind the mirror in all six degrees of freedom (DOF). Therefore the elements for the metrology are chosen to be 2×4 spherical lenses with differ-

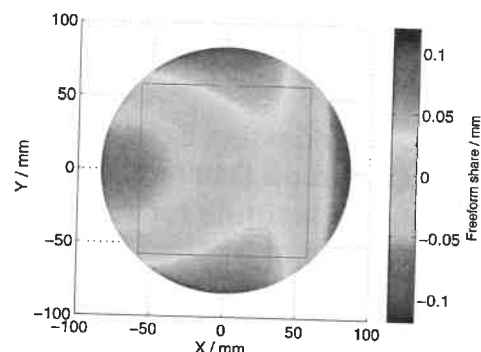


FIGURE 2. Freeform share of an off-axis aspherical mirror with a rectangular CA of $100 \text{ mm} \times 101.7 \text{ mm}$ from an unobscured folded TMA optical design as described in detail by Scheiding et al. [4].

ent sizes on the mirror surface outside of the CA. These elements can be monitored either interferometrically using a computer generated hologram (CGH) in front of a large aperture interferometer or tactile using a high accuracy 3D profilometer. It is beneficial to use the same tool in the same diamond turning process to fabricate the reference elements. But constraints given by the clearance angle of the tool geometry limit the radius of curvature of the spheres to more than 15 mm . Smaller radii of curvature, which are advantageous for a tactile measurement can only be manufactured using a hybrid fabrication process comprising a diamond freeform turning and a diamond milling. Therefore an additional air bearing milling spindle is set up next to the diamond turning tool with sub-micron precision. The numeric control programming and tool radius compensation for different geometries is based on surface normals calculation and a coordinate transformation as shown for a lenslet array by Davis et al. [10]. A major advantage of the milling process is the enhanced design freedom regarding the geometries and their position on the mirror body. For a simplified system integration with reduced alignment efforts, the interfaces have to be retransformed to embody the mirror coordinate system before tilting the mirror as described above. Figure 3 shows a mechanical design drawing of the freeform mirror. The CA and the position of the spherical reference elements on the mirror body are shown on the right hand side. The sectional view shows the vertex of the mirror body, which is on the axis of rotation. The optical axis is normal to the tilted flats for the system in-

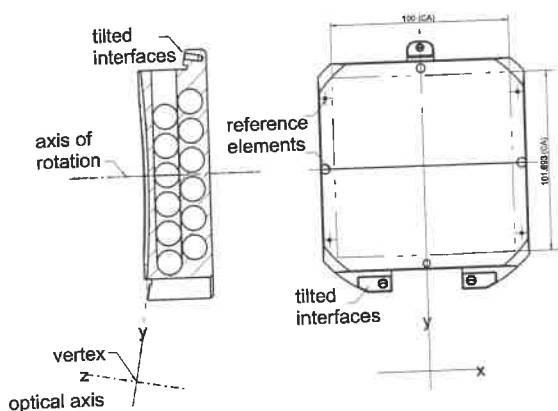


FIGURE 3. Mechanical design drawing of the primary freeform mirror with reference elements for the metrology and tilted interface flats for the system assembly.

tegration. Hence the coordinate transformation is reversed and the system alignment is simplified using these ultra-precision mechanical interfaces with a micrometer registration to the optical freeform surface. The high degree of lightweight about 55% is achieved by removing the material from the area around the neutral plane, where it contributes little to the bending stiffness of the mirror [11].

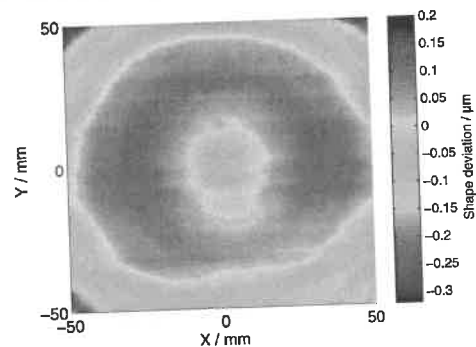
DIAMOND MACHINING AND CORRECTION CYCLE

For the primary freeform mirror investigated in this study, a Nanotech 450 UPL machine with an additional NFTS-6000, from Moore Nanotechnology Systems, LLC is used. The maximum stroke of the FTS is ± 3 mm. The integrated linear scale encoder with sub-nm resolution is in a closed control loop with a voice coil actuator to position the air bearing slide in real time. The NFTS-6000 is able to precisely follow an amplitude of 100 μm at 160 Hz with maximum acceleration of 49.1 m/s^2 [12].

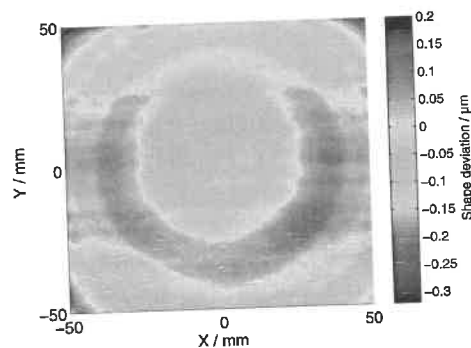
The cutting data are chosen according to state of the art diamond machining parameters. The spindle speed of 250 rpm and a feed of 1.25 mm/min are leading to 5 $\mu\text{m}/\text{rev}$. The theoretical roughness for these cutting data and a diamond tool tip radius of $r_e = 0.5$ mm is 1.9 nm (rms). The cutting depth for the finish machining is 6 μm . The cutting time for a single pass is 70 min.

Upon completion of freeform turning the optical surface and diamond milling the reference elements and interfaces, the mirror is cleaned and inspected to enter the correction cycle. The shape

is measured with a tactile 3D-profilometer UA3P-5 from Panasonic. This measurement device, with a measurement range of 200 mm $\times$ 200 mm $\times$ 45 mm, uses a diamond stylus tool with a tip radius size of 2 μm to scan the surface. The accuracy over the measurement range is 100 nm in X,Y directions, with a repeatability of 50 nm. For slopes up to 30° the accuracy of the installed UA3P is assumed to be below 50 nm.



(a) before correction 87 nm (rms), 423 nm (p-v)



(b) after first correction 49 nm (rms), 275 nm (p-v)

FIGURE 4. Shape deviation from the design freeform surface measured with the 3D profilometer UA3P before and after correction cycle.

Subsequent to the measurement step, the raw data is aligned according to the reference elements and subtracted from the design surface data. The error map is smoothed and inter- and extrapolated. The error map shown in figure 4a is subtracted from the original surface data shown in figure 2. A subsequent tool radius correction and recutting of all surfaces and reference elements leads to an improved surface figure.

RESULTS

The deterministic correction cycle to improve the shape of the off-axis mirror with an on-axis fabrication technology is target-aimed. Figure

4 (a) and (b) shows the shape deviation of the freeform mirror before and after one correction cycle. The original deviation of 87 nm (rms) is reduced to 49 nm (rms) in the CA. A further improvement of the shape seems to be possible with interferometric metrology and further correction cycles. The micro roughness is measured in a field of $140\ \mu\text{m} \times 110\ \mu\text{m}$ using white light interferometry. The error image shows the typical turning marks and the aluminum grain structure with an overall roughness value of 7 nm (rms).

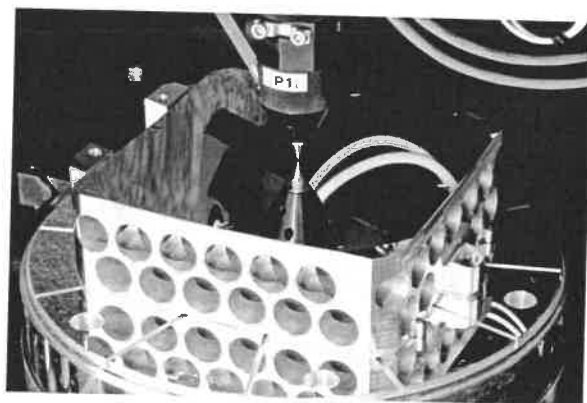


FIGURE 5. Finished freeform mirror with reference elements and tilted interfaces during the shape measurement in the 3D profilometer.

CONCLUSION

This study proves that diamond machining of an off-axis mirror as a freeform mirror in the center of rotation reduces the shape error nearly by the factor two. Error sources coming from the off-axis turning of the mirror, such as temperature induced influences or centrifugal forces can be reduced. Moreover the freeform machining allows the shape error correction of the non-rotationally symmetric error share. The aggressive lightweight structures of the mirror show no influence on the shape deviation. The diamond milled reference elements and mechanical interfaces allow the alignment of the mirror for measurement and integration processes with micrometer precision. Tilted interfaces are used for the re-transformation of the off-axis surface. The shape irregularities in the CA satisfy the Maréchal condition for NIR-MIR-FIR optics starting at a wavelength of $1.4\ \mu\text{m}$. For these spectral ranges the mirror is considered to be diffraction limited.

ACKNOWLEDGMENTS

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MOORE'S LAW, THE CLOUD, THE INTERNET OF THINGS AND PRECISION ENGINEERING

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INTRODUCTION

Secular technology innovations drive precision engineering advances. This was true yesterday and it will be true tomorrow.

The foundational principles of precision engineering were developed over one hundred years ago. The industrial and scientific revolution of the late 1800's developed ruling engines, jig bores, precision milling and turning lathes, optical comparators, and optical interferometers. They utilize the fundamental principles of precision engineering [1] which are taught in several ASPE tutorials

In the last 110 years secular technology advances have driven the precision engineering curve as identified by N. Taniguchi [2] in the mid-1980's. Figure 1 charts some of these secular technology drivers overlaid on Professor Taniguchi's chart.

Technologies which are now commonly found in houses worldwide, such as electricity and air conditioning, where technical drivers to precision engineering in the early part of the 20th century.

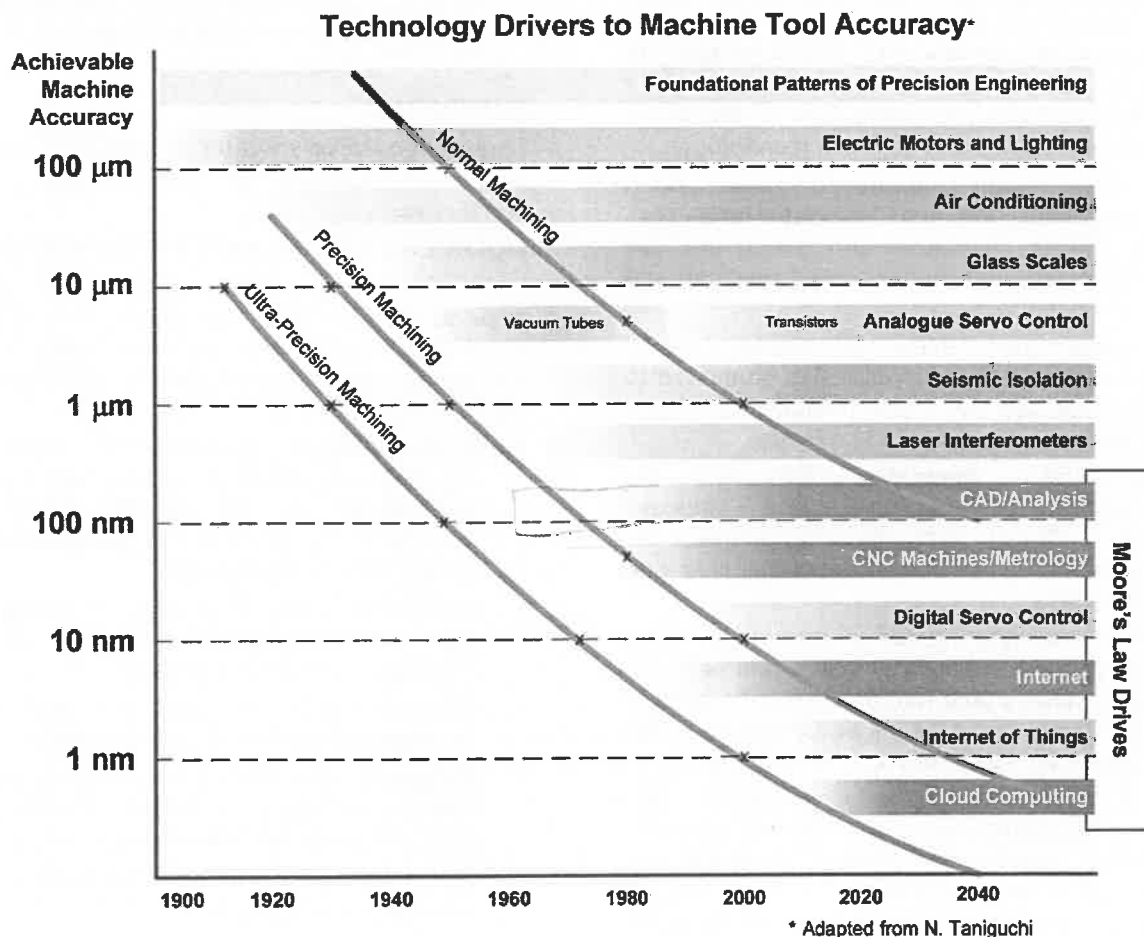


Figure 1: Technology Drivers over 110 Years

SEMICONDUCTORS, ELECTRONICS AND PHOTONICS

Since the 1970's precision engineering has continued its improved accuracy path as driven by new technology. During this time machine tool accuracy improvements have been driven by the invention of the laser, laser interferometers, computer aided design, finite element analysis, analogue and digital servo's, solid-state cameras, computers, high level software languages and seismic isolation systems.

Most of these drivers are outgrowths of the semiconductor and photonics revolutions at the end of the 20th century.

HISTORICAL EXAMPLES

In 1960 the laser was invented. By the early 1970's HP developed the first practical laser distance interferometer. It was driven by innovative laser design and high-speed electronics, enabling stable, easy to use tools. This tool revolutionized machine tool control and calibration with high resolution AND the ability to minimize abbe offset. It would not have been possible except for the invention of the laser, modern electronics and software technology.

In 1972 Intel introduced the 8008 microprocessor, the first "computer-on-a-chip". These enabled distributed processing and low cost computers that revolutionized machine and instrument control.

Throughout the 1970's to 1990's analog electronics for motion control also continued to develop, and are now being displaced by digital servo controllers and control software.

Lower noise electronics improved the performance of capacitive and inductive sensors. These improvements enabled the development of higher precision spindles and machine ways.

Surface measuring interferometers for form and roughness were enabled by faster data analysis, improved micro-stage motion, proximity sensors and solid-state cameras, all an outgrowth of the semiconductor and photonics revolution.

In all these cases semiconductors and electronics were the key drivers towards improved performance whose computational speed gains were explained by Moore's Law. For precision engineering in the last 40 years Moore's law has been one of the most influential drivers for machine tool accuracy improvements.

THE COMPUTING POWER AFFECT

In summary computing power has driven improvements in precision engineering in at least five ways:

- **Accuracy:** With repeatability, systematic errors can be mapped and corrected in real time improving process uncertainty
- **Machining capability:** Fast tool servos machine shapes with spatial frequency ranges undreamt of 20 years ago
- **Environment:** Thermal stability through intelligent controllers that integrate sensors and HVAC to achieve orders of magnitude improvements.
- **Simulation:** Entire systems can be designed and tested before manufacture, shortening the learning curve and improving performance
- **Synergy:** Each of these improvements build on themselves to produce more repeatability, fancier machines, better lenses, better computers; bootstrapping themselves to more precise tools.

THE FUTURE

Three future drivers will be considered: Moore's Law, the Internet of Things, and cloud computing.

MOORE'S LAW: 10X FASTER IN 5 YEARS

Intel's Paul Otellini notes Moore's law [3] is assured for the next five years, that means computing power will be ten-times faster or ten-times less expensive at today's performance. Possible benefits are.

- **Environmental Insensitivity:** Machines that sense vibration and servo fast enough to counter act its affects during the manufacturing process. If the mechanics can be made stiff enough!
- **In Situ/Real Time Metrology:** Metrology for form and roughness in real time, on the machine – a trend in its infancy.
- **Design:** Better machine and instrument designs due to more complex models
- **Materials:** New materials developed due to better, more complex models of alloys and composites for specialized properties
- **Fool Proof Controls:** Machine controls that catch and correct operator errors
- **Expert Systems:** Upon entering part parameters, the machine or instrument will

automatically set up and instruct the user on the next sequence of events to perform.

- **Maxwell Equation Level Corrections:** Coherence Scanning Interferometers (CSI) measure mid and high spatial surface frequencies. Correcting limitations in these systems such as thin film and edge effects, rough surface and vibration, and phase change on reflection involve complex electro-optical solutions. Increased computational power will allow these limitations to be measured and corrected for more consistent and user-friendly results. Also new measurements might be possible such as measuring the sides of walls [4].

CLOUD COMPUTING

"Cloud computing is a model for enabling ubiquitous, convenient, on-demand network access to a shared pool of configurable computing resources..." [5]. Possible benefits are.

- **Metrology Algorithms:** Cloud computing could enable uniformity of analysis code. Standardized algorithms could be accessed from Standards Laboratories (NIST, NPL, PTB, etc.). No longer would data sets be analyzed differently.
- **Traceability:** National Standards laboratories could develop certification processes. These processes, when run in the field, could be monitored and analyzed at the Standards laboratory. The results would be certified as traceable without the need for transfer artifacts.

THE INTERNET OF THINGS

Internet users are globally growing at 400% per year and Internet speed is increasing at 50% CAGR as shown in Figure 2.

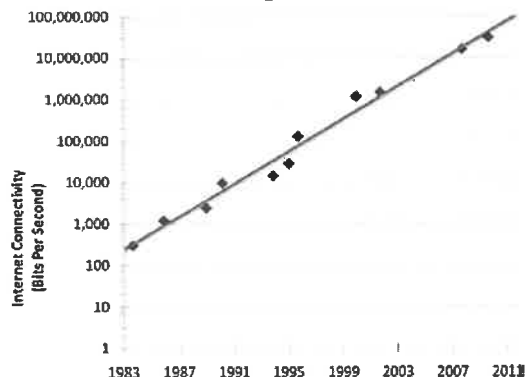


Figure 2: Internet Speed Growth Rate

Leveraging Internet growth will be the Internet of Things, a new technology that promises to drive precision engineering. "[The] Internet of Things, [are] sensors and actuators embedded in physical objects—from roadways to pacemakers ... linked through wired and wireless networks, often using the same Internet Protocol (IP) that connects the Internet. These networks churn out huge volumes of data that flow to computers for analysis. When objects can both sense the environment and communicate, they become tools for understanding complexity and responding to it swiftly. [6]

Considering the growth of the internet shown in Figure 2 and adding a network of interconnected – analyzing sensors the Internet of Things can be a key driver to machine tools accuracy improvement. For example:

- **Known Measurement Uncertainty:** Uc, is a mystery to most users today. Combining the input of multiple environmental sensors, system error mapping (including the actual part begin manufactured or under test), error budgeting plus historic calibration, the product or result could include an Uc result.
- **Total Real-Time Corrections:** Consider a coordinate measuring machine (CMM) with hundreds temperature, pressure, strain, distance, angle, and weight sensors integrated throughout its structure and a computer program rapidly responding to correct for systematic and random errors to provide nanometer metrology on a factory floor ... at low cost. Also the manufacturer could have real time data on all their installations to gain even deeper understanding of the system dynamics so they could offer performance upgrades by remote analysis and a software download. This was a vision in the 1980's [7]. The Internet of Things can make this vision possible.

WHAT ELSE IS NEXT?

Beyond computing power, cloud computing, and the Internet of Things what other technologies offer new opportunities.

Not addressed here is the photonics revolution. The laser has continued to evolve as have solid-state lighting, displays, imaging systems, solid-state cameras with ever wider spectral sensitivity, and optical computers. Add to these data storage capabilities, Tablet Computers, the wireless networks with unlimited bandwidth,

designer materials, and advanced software programming and other technologies not identified here.

There are new technologies to explore and apply, and new products to be built; the leaders of the future will master these technologies and continue to push precision engineering and even normal machining to greater levels of accuracy.

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WEB-BASED DRAFT INSPECTION SYSTEM FOR DESIGN AND FABRICATION OF PRECISION MOLDS

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ABSTRACT

A Web-based CAD viewer (WebCMM) with draft examination function for design and manufacture of injection molds is studied. In the distributed environment, this system transforms CAD data into lightweight files through ACIS kernels and InterOp on the translation server. The draft examination function has been constructed through ActiveX using visual C++ and OpenGL.

KEYWORDS: CAD viewer, Collaboration, Ejection draft, Injection mold, Internet browser, Undercut.

INTRODUCTION

Followings are required in manufacturing of injection-molded products: 1) reduction of design and fabrication time, 2) improvement of product accuracy and quality, and 3) prompt response to design change. Mold companies are using 3D CAD/CAM systems to fulfill the requirements. However, as the CAD model of an injection-molded product does not reflect all requirements of the mold design or injection process, various factors should be examined prior to design of molds. These include product configuration, ejection draft, undercut extraction, contraction ratio, gate/ejector location, etc. The undercut extraction is especially important in selecting parting lines and faces, as well as in creating cores and cavities [1-2]. Thus various review sessions are indispensable for the mold company, clients, and related companies. Many meetings are held between departments and companies to derive proper design results. As the companies are often located in distant locations, these meetings cost time and money. Currently, most mold companies indicate markups on fax or email messages and 2D drawings, and exchange these documents instead of holding additional meetings. This degrades design quality.

In addition, in small or medium size mold companies, as the 3D CAD/CAM systems are expensive and difficult to use, they are mainly used by design experts. Technicians in the manufacturing department use 2D CAD drawings

for drilling, milling and assembly. This requires transformation of 3D data to 2D drawings by the expert. It delays lead-time. The 2D drawings degrade quality of precision jobs in the mold fabrication too. Therefore, most injection mold companies want to use collaborative Web viewers in the processes [3-5]. In this paper, an ejection draft examination system to verify undercuts even in complex shapes with free-form surfaces is studied according to the parting direction specified by an expert mold CAD designer on the previously developed web-based CAD viewer [3-5].

DESIGN METHOD

For example, undercuts require side-cores, side-cavities, pins, etc. They increase cost and delay mold production. Ejection draft is indispensable for secure fabrication of molds. Chen *et al.* [1] carried out a study for selecting the parting direction to minimize number of undercuts. Ganter and Skolgund [2] conducted a study on the recognition of undercuts through the boundary representation model (B-rep model) of a product. However, this algorithm recognizes only concave undercuts. In this paper, an ejection draft check system to verify undercuts even in complex shapes with free-form surfaces is studied according to the parting direction specified by an expert mold designer through the Web viewer as shown in Fig. 1. Its design methods are as follows: First, upon selection of an injection company, the client registers CAD data of the injection-molding product to the collaborative system server. The injection company then checks the product configuration and selects an appropriate molding company. Subsequently, the selected molding company's design department downloads the product data from the server and designs the corresponding mold. Upon completion of the mold design by the expert mold CAD designer, the design data is registered on the server and the design review is requested through the CAD viewer [3-5]. Engineers and technicians of related departments review the design on the server, and look into problems associated with mold manufacturing, mass production, ejection

draft, undercuts, etc. If discussion regarding the location of the parting line according to undercuts or magnitude of ejection draft is necessary, the injection and client companies contact for review. They can then access the collaborative system server to check the 3D configuration as well as the parting line, ejection draft, and undercuts. If problems such as the flash occurrence or abnormal product configuration are found, design modification and review are requested. These processes are repeated until the parties arrive at the final agreement. 3D configuration of the CAD viewer allows participants to conduct a more intuitive review than 2D models, which allows everyone to respond promptly and accurately to any design changes.

System Architecture

Fig. 2 shows the Web-based client-server structure composed of the designer client, the integral server and the Web client. Client and server data are exchanged through socket communication [3-5]. The Web server provides the user access Web page and enables corresponding authority. It also enables the designer to register drawings on the Web page. The design verifier searches the design database on the Internet and examines the design and ejection draft. The searched data is transferred from the server to the client to allow examination. Examinations are conducted on the retrieved design data over the Web, and the result is saved as a lightweight file, which is sent to the integral server to be shared among multiple users.

Client CAD Viewer

Fig. 3 shows the client CAD viewer composed of the lightweight file reader/writer, the 3D viewer, measurement, markup and markup read/write modules [3-5]. Each module is created through ActiveX technology of Microsoft. HTML is used to notify the ActiveX controls on the Web, and VBScript is used for plug-in interfaces. In this architecture, ActiveX is automatically installed on the client's PC when a client is connected to the server. After the first installation of ActiveX on the client's computer, versions of the distributed ActiveX controls are checked during every login process, and updated ActiveX controls are automatically distributed to the client. The draft examination process is performed on the client's PC, which reduces network dependence and server overload during 3D visualization and/or draft examination of injection molds. In addition, as the client CAD viewer does not have a CAD translator and its file size is around 519 KB, this

system enables clients to verify the mold design and draft examination results on the Internet explorer rapidly.

CAD Data Interface

Commercial CAD softwares express CAD models with parametric surfaces. However, computer graphics environment does not directly allow visualization of them on the screen. To visualize CAD models, we use a tessellated triangle mesh. For the Internet visualization, we generate the lightweight data from the CAD data through InterOp translator using the ACIS CAD kernel. InterOp let you access 3D data without extensive code modification, expensive CAD software or APIs (application procedural interface). It converts commercial and neutral CAD data into ACIS kernel data transparently [5].

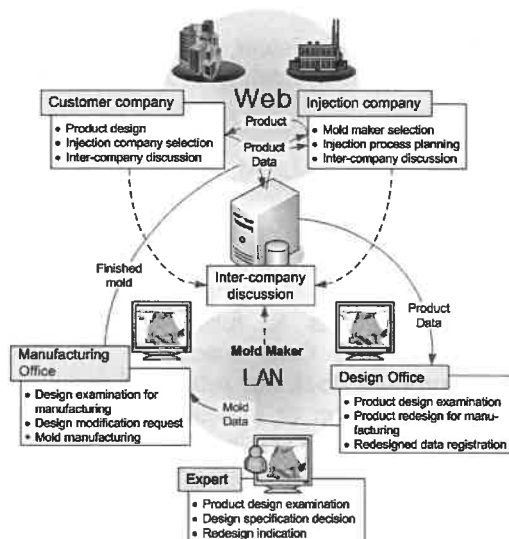


FIGURE 1. New collaborative framework for injection mold design.

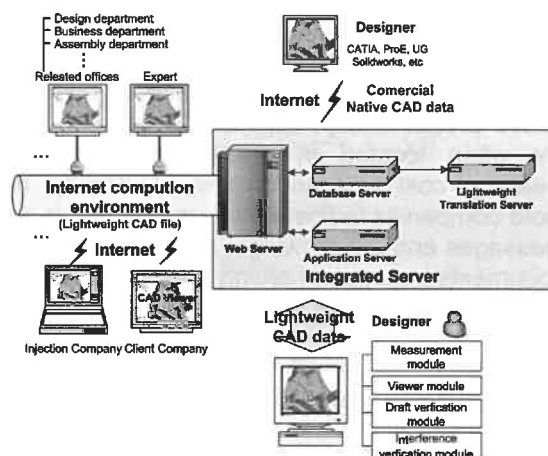


FIGURE 2. Framework of the Web-based CAD viewer (WebCMM) for injection mold design.

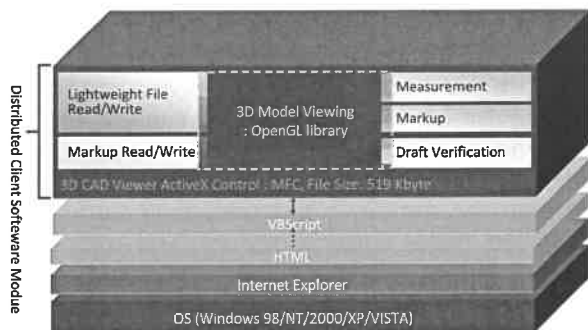


FIGURE 3. Structure of the CAD viewer (WebCMM) at the client side.

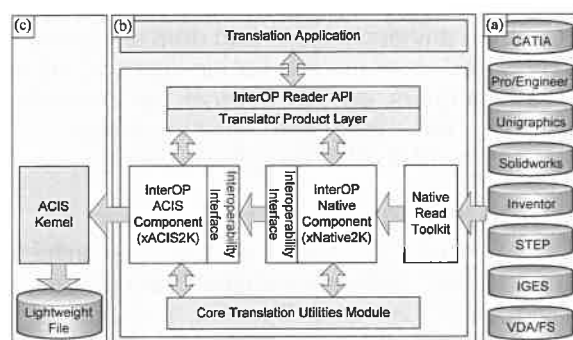


FIGURE 4. Structure of the lightweight file translator for CAD data interface.

Extraction of Geometric Data

Fig. 4 shows the lightweight file translator. (a) displays CAD files produced from commercial CAD systems. (b) represents InterOp Reader of Spatial Co., which converts CAD files into the ACIS kernel. (c) describes the 3D modeling ACIS kernel, which generates lightweight files [5].

Recognition of Undercuts

Experts intuitively find undercuts of an injection-molding product with simple configuration and then select the parting line. But for a complex product, usage of commercial CAD software is indispensable to find undercuts. However, it is very expensive and requires large-size CAD files which make them inappropriate for the distributed environment on the Internet.

Parting Procedure

In general, a molded product is ejected to the opening direction of an injection machine. Holes or protrusions of the molding product hinder ejection in the specified direction. They are classified into internal and external undercuts according to their locations in the molding product. Fig. 5 shows mold design processes to create cores and cavities. Designers derive the ejection draft and injection conditions based on CAD data.

After the parting direction is decided upon, the parting line is selected according to the exterior of the molded product and post-processing methods. Finally, the parting face is created according to the location of the parting line, and the design department commences the actual mold design process including the core, cavity, and side core.

Examination Algorithm

Our draft examination is implemented through the vertex data of the lightweight file and the normal vector of each vertex. The draft examination algorithm is as follows: First, the parting direction vector and the triangular patch normal vector are defined as N_{PD} and N_{Face} for draft calculation as shown in Fig. 6. An angle in the direction perpendicular to the parting surface, which allows ejection of the molded product without deformation, is called the ejection draft, and is calculated according to Eqs. (1)~(3).

$$N_{PD} \cdot N_{Face} = \|N_{PD}\| \|N_{Face}\| \cos \theta \quad (1)$$

$$\theta = \cos^{-1} \frac{N_{PD} \cdot N_{Face}}{\|N_{PD}\| \|N_{Face}\|} \quad (2)$$

$$\theta_{Draft} = -\left(\theta - \frac{\pi}{2}\right) \quad (3)$$

Where θ is the angle between the parting direction vector and the triangular patch normal vector, and has a value between 0 and π . The ejection draft is perpendicular to the parting direction, and parallel to the normal vector of the side surface of the mold base. The calculated θ is therefore substituted into Eq. (3) to obtain the desired draft, θ_{Draft} . Then, the calculated draft has a value between $-\pi/2$ and $\pi/2$.

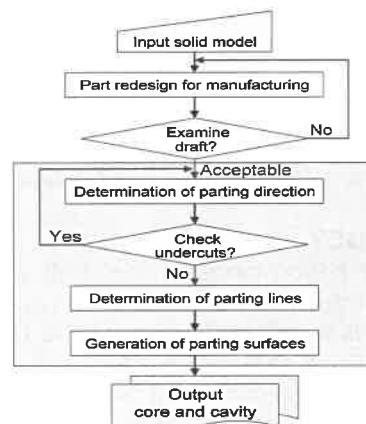


FIGURE 5. Design procedure of parting direction and parting lines.

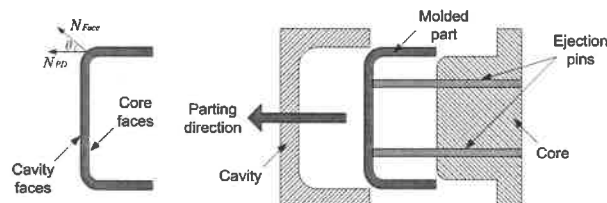


FIGURE 6. Ejection process of a molded part.

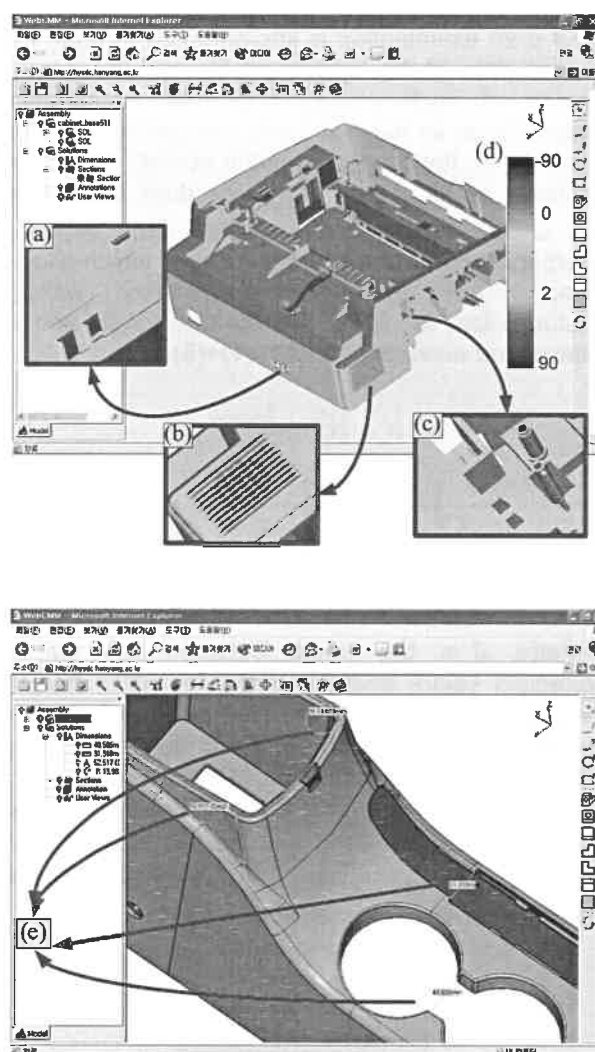


FIGURE 7. Draft examination of a printer and dimensional check of a center console on the CAD viewer (WebCMM) over the Internet.

CASE STUDY

We verify performance of the draft examination system through CAD files. It shows computed draft results in different colors on the CAD Viewer (WebCMM) so that users can easily search for undercuts and ejection draft over the Internet. Fig. 7(d) shows the color scale to distinguish amount of draft angle. Undercut areas are displayed in red and other areas in blue. Draft examination using

the developed system indicates that the overall ejection draft of the printer part is about 2°. Fig. 7(a), (b), and (c) show undercuts occurring on the part. Based on the examination results, collaborators are able to select a new ejection direction or to request modification of the mold design for removing the undercuts through the addition of side cores or cavities. The lower part of Fig. 7 shows a center console of a car. Various dimensional verification functions shown in Fig. 7(e) confirm design results in 3D domain. They enable to substitute the developed CAD viewer (WebCMM) for 2D drawings during the fabrication of molds. Contrary to 2D paper drawings, it measures any dimensions and draft angles easily. Clients can communicate design ideas or opinions to collaborators on the Internet conveniently. It reduces the lead-time and increases the productivity of mold making processes.

CONCLUSIONS

A Web-based CAD viewer with draft examination function has been constructed through ActiveX using visual C++ and OpenGL. It expedites the injection mold design process and reduces lead-time to fabricate molds, as well as improve accuracy of molds and quickly responds to design changes.

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AN INEXPENSIVE DEVICE TO TEST CLIMATE CONTROL SYSTEMS IN A MANUFACTURING ENVIRONMENT

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Having been involved in manufacturing for the last 49 years, my journey into the world of precision has brought me to the conclusion that 'the pursuit of precision' involves three things.

'passion' the demon that keeps you at work late into the night to try one more thing.

'money' never enough.

'intellect' to figure out the next best course of action.

You can always make up for a short coming in one by using the others, (I think most of us have gone 'dumpster diving' to find something that will work because we haven't got the money or time to buy something off the shelf). All of us here have an "I need this" streak.

In that vein this item was constructed.

The project that led to this device required a pattern of close tolerance holes to be placed on two datum surfaces, and one opposing surface that were referenced off of a third datum plane. The tolerance for location of 6 holes was 8 microns true position in a 5" x 24" x 36" (125mm x 610mm x 915mm) area.

The major challenges resulted from the different thermal characteristics of a large thin-walled part compared to the machine tool (the difference caused by 1 degree represented the tolerance), the part's tendency to give and take temperature rapidly and from the need to run the part at different heights on the machine (some operations at normal table height while other operations were at 6 feet (1.8m) off the floor) that created issues of temperature stratification.

The initial trials revealed that the part's tendency to give up a temperature 'soak' (the action of stabilizing the temperature of an item over a length of time, typically 24 hours, with climate and thermal masses) and take on the operator's body heat were issues that had to be addressed. The climate control system in operation was able to hold +/- .25 degree in a 'soak' with large

steel parts but the variation in the moving air temperature (sometimes over 1 degree) kept this part in constant thermal motion.

Attempts to monitor the air temperature and adjust the climate control system resulted in inconsistent results and use of an end standard made out of a similar material to the part proved to be repeatable but not practical or cost effective.

The money solution called for a new climate system.

The lack of such funds led to the creation of this 'test bed' to find a way to allow that part to take and hold a 'soak'.

A large cast iron v block (3" x 4" x 18", 75mm x 100mm x 460mm) was used to represent the machine tool and a standard (of the other material) was made to represent the part (1" x 2" x 17", 25mm x 50mm x 430mm). The standard was elevated in the v with a series of precision balls to provide free motion with respect to the cast iron section and provide a large enough gap to minimize the temperature transfer between the v block and the standard. One end of v had a 5 μ inch (.00012 mm) micrometer head to provide adjustment. The other end was raised to provide a slight angle (3.5 degree) and fitted with a 2 μ (.00005mm) mikrokator. (Figure 1)

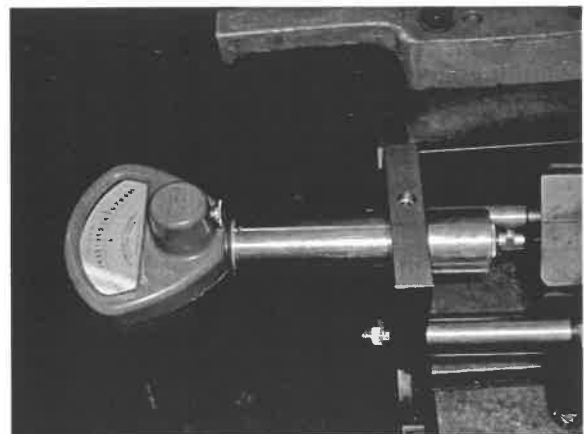


Figure 1

The advantages over a typical vertical arrangement on a surface plate were many. First, there was no 'house of cards' issue about the whole arrangement being knocked over. Second, there was no transfer of temperature from the surface plate to the gage. Third, the device could be placed anywhere in the shop for any length of time since a surface plate was not required. Finally, the device was able to be scaled to a larger size to increase sensitivity.

The balls (3 on one face and 2 on the other) were supported at the desired height with spacers. I found that by using spacers that were approximately 1/4 the diameter of the ball, sensitivity was greatly increased. As a final check a ball was placed between the micrometer and the end standard. If the part didn't move freely at all times the ball would drop out. (Figure 2 and 3)

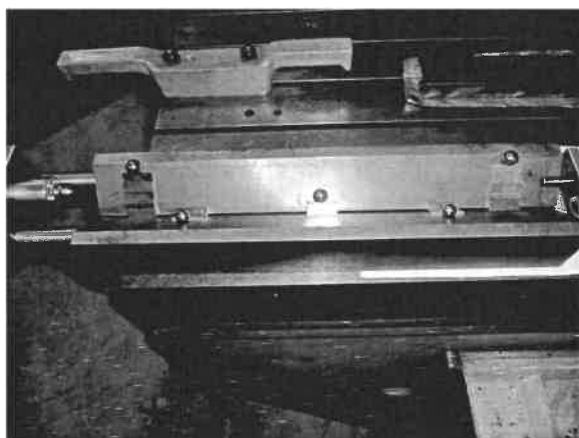


Figure 2

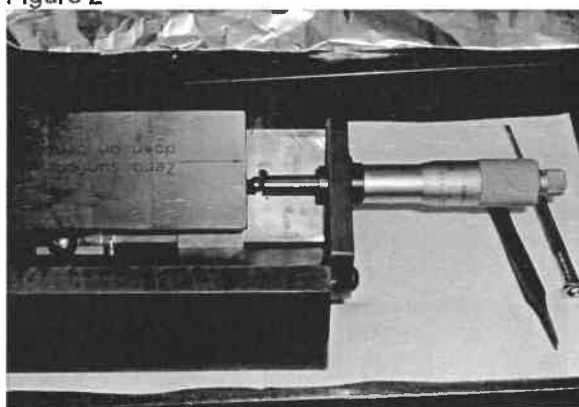


Figure 3

The test device was placed in close proximity to the machine tool to be used and allowed to

stabilize for 24 hours. Readings were then collected over a 2 day period. The data indicated a constant movement in a 30μ inch ($.0008\text{mm}$) range. Once this baseline had been established 2 primary means of reducing this movement were to be tested, coating and covering. Coating involved several types of plastic, covering involved aluminum foil in layers. At this time the device was moved to a more convenient location for this testing to begin and entirely new set of data presented itself prior to testing the coverings.

In the new location the movement over a two day period was 90μ ($.0022\text{mm}$) inch. At this time 'passion' kicked in and the device designed to be used to test coatings (plastics, cork) and coverings (various foils) was used for a detailed analysis of the climate control system. Over the next 30 days the device was placed in eight locations in the precision area (2400 square feet) including a final return to the original location. The data revealed numerous 'micro climates' over the shop floor with the best areas being slightly less than 30μ movement and the worst to be over 100μ . The data also showed an extended period of stability at all locations (less than 10 to 20μ) over an extended period of time (5-6 hours) while the room was being actively worked in.

The construction of the room (mostly thick stone) and the presence of operators, lights and active machines added enough heat that the climate system didn't activate very often. When it did the measurement extremes returned and took approx 1-2 hours to level out.

These results led to a relocation of the 'soak area' normally used and the following process being followed. The part was located on the machine the prior evening with an end standard mounted and covered with several layers of Aluminum foil. In the morning the climate system was checked for cycle and forced into operation if necessary and then turned off. The next 1-2 hours were spent verifying the parts location and machine zeros with the final operation being a referencing to a standard of similar material to the part.

Compensation numbers were recorded and used for that day's operations and then the part was set up for the next day's operations. Processing and inspection time involved 4 days and represented no additional time over the original test parts. Over a dozen of these parts

have been run with this process over period of several years and the compensation numbers have remained so constant that the standard referencing is only now done at final inspection with the total variation over these parts being less than 3 microns true position.

I feel that this is a good example of 'using what we have to get the job done' and an excellent way to test a climate system. The "Crusade" of any climate control project is the desired temperature (20°C in this case), however the "Holy Grail" of any system should be stability. I found this an excellent device to identify the "sweet spots" and problem areas.

If needed, an even more sensitive test stand could be made by using Aluminum for the v-section and Invar or Zerodur for the reference standard and also replacing the Mikrokator with electronic gages. This would result in 4-5 times increased sensitivity without increasing the size of the system.

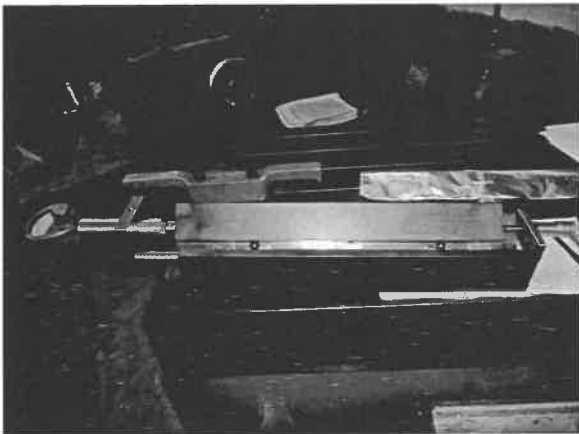


Figure 4

A STUDY OF INFLUENCE OF LONG-PERIOD GROUND MOTION ON SEMICONDUCTOR INDUSTRY.

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INSTRUCTIONS

In the field of precise positioning and precise measurement, active isolation system is generally used. Especially, it is indispensable unit in the system that demand high accurate isolation performance from outside in the semiconductor device.

By the way, accuracy is also severe with the latest semiconductor lithograph machine used on all over the world. And it is extremely expensive device that does billions of yen for each. On the other hand, The building of the semiconductor firm which sets up the lithograph machine cost several hundred million dollars. Here, it introduces a recent huge earthquake. As for it, a Sumatra earthquake of Indonesia, a Sichuan large scale earthquake of China, etc. In these huge earthquakes, immediate damage is reported in the earthquake occurrence point in the Sumatra earthquake. The seismic wave becomes a long-period ground motion. And the influence reached the entire earth. In addition, the long-period ground motion influences the productivity of semiconductor firm. For instance, the machine error occurs in lithograph machine of the Asian region. Various countermeasures have been put to practical use for the earthquake usually, such as aseismatic mechanism is installed. However, machine error occurs for a long time of two hours from one hour in case of long-period ground motion. Moreover, the device breakdown doesn't occur after passing the earthquake. And, concrete countermeasures are not adapted, because it returns usually to the state of operation. In Sichuan earthquake in China, the damage of epicenter was extensive. Huge energy of earthquake causes a long-period ground motion. And, it turned several times round the earth surface. This is observed real data. The long-period ground motion reached Japan where 2,000-3,000 kilometer left the epicenter several hours later. And whole land in Japan vibrated in the low frequency. Especially, the semiconductor firm had been occasionally overcrowded in the west Japan. The machine

error happened frequently. And, productivity was remarkably decreased for several hours. For the semiconductor firm, the utilization rates of the device is managed extremely severely. The stop of only ten minutes might be counted as a breakdown. In such a situation, economic loss of the stoppage in production of several hours in many firms are also large. Here, based on such a background, as a long-period ground motion countermeasures, first of all, it explained the feature of the long-period ground motion that becomes a problem and the behavior of the semiconductor lithograph machine. In addition, it explains the sensor specification and the building characteristic, etc. to evade this phenomena.

Entire overview of main body

Figure 1 shows the general view of the semiconductor lithograph machine. This machine sets up on the installation floor that is called a pedestal which is in the base of the machine. Generally speaking, Projection lens is mounted on the main body based on isolation device, that is air-spring. Reticle stage is installed on the main body, and wafer stage is hang over from the main body as shown in Figure 1.

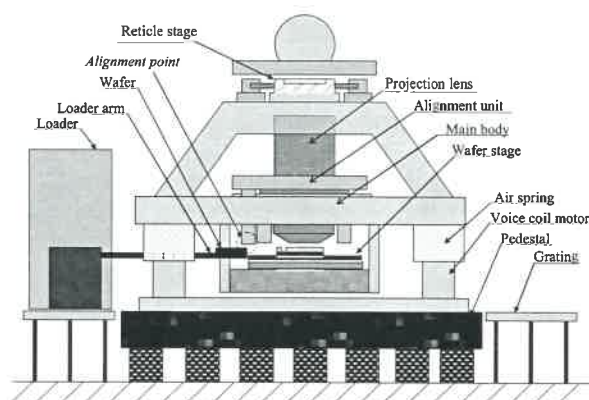


FIGURE 1. Main apparatus of semiconductor lithograph machine.

Voice coil motor is main actuator of the lithograph machine(hereafter, abbreviation as VCM)and an air-spring are mounted on the base of the main body of the machine. In addition, the projection lens, optics system of the alignment etc. are mounted on the main body of the lithograph machine. Thus, this main body is isolated from the floor vibration with the air-spring , and the isolation performance greatly related to the stiffness of the air-spring, that is , the capacity of the volume of the air-spring. In general, main body is supported on the pedestal through three air-spring support legs. In each supporting legs of the machine, it provides with three sensors that measure a pedestal and a relative position with the main body respectively in the vertical direction. In addition, to measure the position of the main body from the floor, there are 3(6 in total with vertical direction) positioning sensors for horizontal direction. The position sensing device of the main body from the pedestal are adjacent and arranged with three air-springs. In similarly, three accelerometers are mounted in the vertical direction, another three are mounted horizontally, and the movement of six degrees-of-freedom is measured. Moreover three VCM are horizontally and vertically arranged.

When the wafer is loaded, it has made the alignment at the alignment point in the Figure 1. The clearance at the alignment point has tolerance. It is less than 1mm on the current machine. The machine can be assembled easily due to professional field engineer, however, the machine has some error under the following phenomena.

Long-period ground motion

As the above-mentioned, huge earthquake, Sumatra, Sichuan, has a big power and big energy, and it caused a long-period ground motion, passed through on the surface of the earth to all over the world, including Europe, Asia, US, on several times. In Japan, there are many troubles on due to long-period ground motion on the semiconductor firm. There is a stratum relating to how to vibrate. Figure2 shows how to transfer ground motion to all over the world.

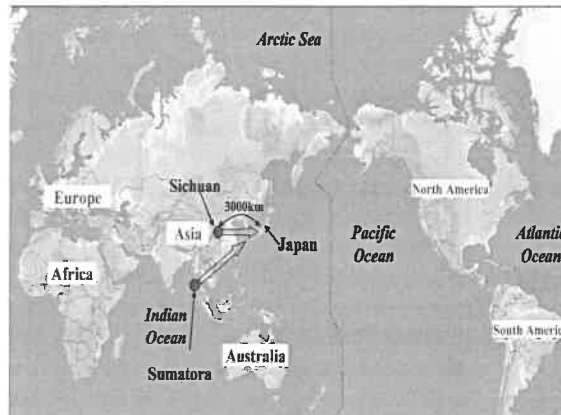


FIGURE 2. How to transfer to the world.

Table 1. shows the feature of long-period ground motion, such as Seismic level, frequency, maximum displacement.

TABLE 1. The table of feature of long-period ground motion.

Item	Content
Seismic level	1 or less
Frequency	1.5Hz or less
Maximum displacement	0.5mm or more

Figure3 shows a real vibration data of long-period ground motion observed in Japan in case of Sichuan in China. According to the figure3, the ground vibrates in all of direction, that is x, y, and z. Its amplitude is $\pm 500 \mu m$ in all direction. The main frequency of ground motion is about 0.1~0.2Hz in all direction.

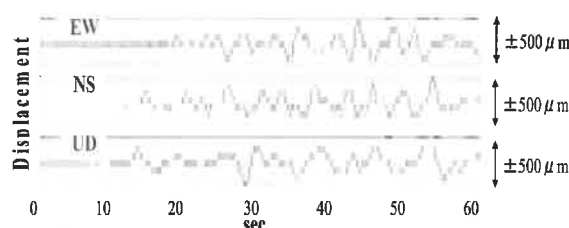


FIGURE 3. The influence of China Sichuan large scale earthquake observed in Japan.

The relationship between vibration of machine and ground motion

Figure 4 shows the vibration at the alignment position of the semiconductor lithograph machine in the Figure 1. The data is calculated by real data of Figure 3. Showing as it does in the Figure 4, the position at the alignment point is over the tolerance of the machine, the machine has trouble under this condition.

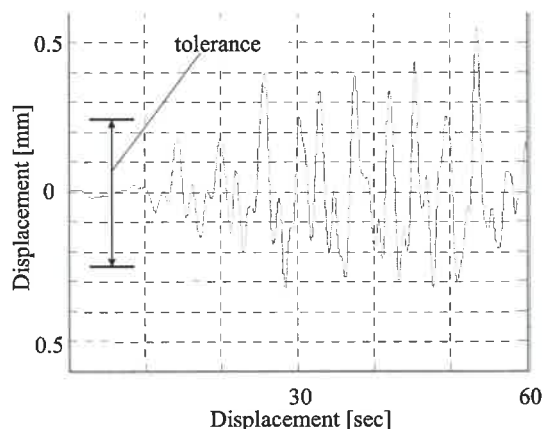


FIGURE 4. The influence of China Sichuan large scale earthquake observed in Japan.

The relationship between stratum and natural frequency of the ground motion

As for the ground motion, it is relating to its stratum characteristic too much. Because its stratum has a natural frequency of stratum. Figure 5 shows the stratum of Japan. The red of right-side of Figure 5 means the stratum with 6-10 seconds natural period. Here, we will focus on the cyan of left-side of Figure 5, we can understand the same distribution as red color of right-side. It follows that we can understand it by comparison the stratum with ground model of Japan in the Figure 5. that is ,the ground with the same stratum has the same natural frequency.

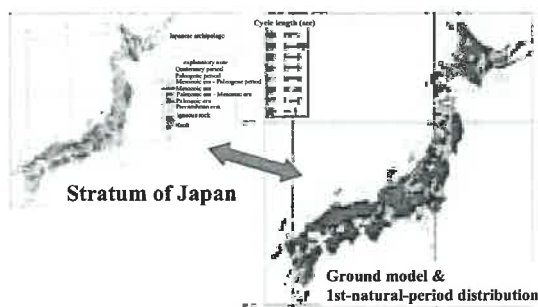


FIGURE 5. Stratum of Japan.

Figure 6 shows the stratum of east United States. By here, we have explained the issue on Japan. The same things can be said even in the United States, that is , there are a little many error on the semiconductor lithograph machine, some of them are unknown error due to long-period ground motion.



FIGURE 6. Stratum of east United States.

Conclusion

Huge earthquake causes a long-period ground motion, and it transfer to very far away, almost all over the world. This ground motion doesn't destroy the machine but causes the servo error. This motion has long sysle, therefore it decreases the utilization rates of the machine. We must establish countermeasure to long-period ground motion because it cause a lot of economic damage to semiconductor industry in the future.

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Deterministic Fabrication of NIC Targets

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ABSTRACT

The National Ignition Facility was dedicated on May 29, 2009 and its 192 laser beams are now engaged in daily science experiments supporting campaigns in inertial confinement fusion and high energy density science. Key focus areas of current experiments include exploring the dependencies on hohlraum drive and symmetry, and thus depend on a reliable stream of ignition-like targets with repeatable properties and dimensions and a comprehensive metrology characterization. The ignition target is a precision assembly comprising a hohlraum, capsule, and "thermal-mechanical package" (TMP) designed to provide kinematic location of components and thermal uniformity under cryogenic conditions.[1][2] During the past year, the fabrication of targets has evolved to a greater level of automation and determinism. Error budgets supporting the fabrication and validation of critical assemblies have been formulated and are now being validated with metrology of completed targets. As an example, the specification band for hohlraum length is determined by a tolerance stack-up of several component length tolerances, gaps due bonding operations, waviness of mating surfaces, and non-repeatability in assembly, and metrology uncertainty. We have systematically revised the design of the TMP embodiment of the target with a greater focus on design for manufacturability and measurability. Different algorithms for combining errors have been considered in assessing the likelihood of meeting physics specifications. A data archive is now being formulated and mined for feedback into the design and fabrication processes, with continuous improvement in metrology uncertainty. While meeting the needs of repeatability and predictability in fabrication, a suite of new assembly tools provides increased

throughput while offering agility in accommodating varying size scales and novel target features. The goal is to provide the breadth of scale and geometry variations required for upcoming shot campaigns, while continuing to improve throughput, yield, and precision, consistent with ignition specifications.

AUSPICES and REFERENCES

This work was performed under the auspices of the US Department of Energy under Contract DE-AC52-07NA27344. LLNL-ABS-420484

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FUNDAMENTAL STUDY OF COATING USING ELECTRICALLY POWERED SLIDERS

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INTRODUCTION

Surfaces of many engineering products, such as cars, are coated. Coating is important since it improves visual and tactile quality and has preservative effects of the products [1]. However, there are not clear condition reports such as coating injection rate or coating times for products with various types of surfaces. In this work, we developed a trial coating robot and evaluated a relation between coating characteristics and surface textures for high quality coating to products.

OUR DEVELOPED COATING ROBOT

Figure 1 and table 1 show an overview and the specification of our coating robot respectively. Our coating robot is composed of the two electrically powered sliders along X and Y axes and a spray gun installed on the X axis. The speed of the sliders and the ON/OFF switch of the spray gun can be controlled by a computer. In our experiments, we performed three types of coatings; under coating, top coating, and clear coating, on metal sheets. We jetted each coating in one minute and then calculated injection rate from the weight change of coatings. We set air pressure 0.3MPa and coating speed 50mm/s as a coating condition in our experiments.

DIFFERENCE OF SURFACE ROUGHNESS OF METAL SHEETS BY COATING TIMES

For evaluating the difference of coating quality by changing the number of coating operations, we coated metal sheets several times by our coating robot and measured the coated surfaces with a scanning white light interferometer. Our coating process was performed in the order of degreasing, under coating, top coating, and clear coating. We executed under coating N_u times and top coating N_t times to 16 sheets with a different number of executions N_u and N_t , where $0 \leq N_u \leq 3$ and $1 \leq N_t \leq 4$. We also performed clear coating twice to all the sheets.

Figure 2 shows the metal sheets after coatings. As a result of our experiment, the most smooth coated surface was obtained when $N_u=2$ and $N_t=2$. Figure 3 shows the measurement result of surface roughness before and after coatings with $N_u=2$ and $N_t=2$. The root mean square value (Sq) of the sheet was reduced from 1.28 μ m to 0.44 μ m and it shows that much smoother surface was obtained after coating.

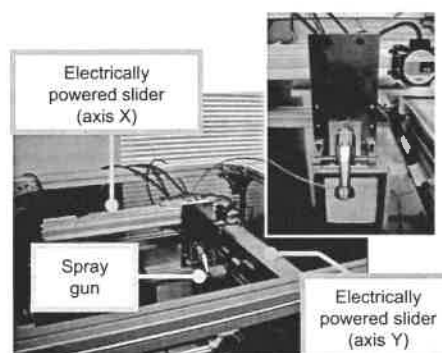


Figure 1. An overview of our coating robot

Table 1. Specification of our coating robot

Range of Coating	Slider axis X: 400 mm Slider axis Y: 350 mm
Maximum speed of coating	200 mm/s
Injection rate	Under coating : 4.5g/min Top coating : 3.0g/min Clear coating : 3.0g/min

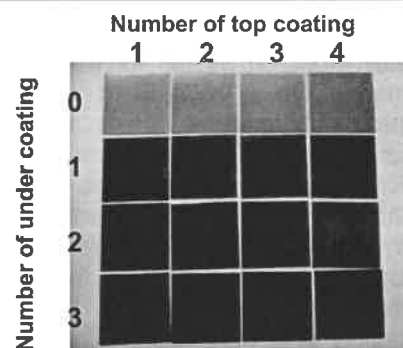


Figure 2. Metal sheets after coatings.

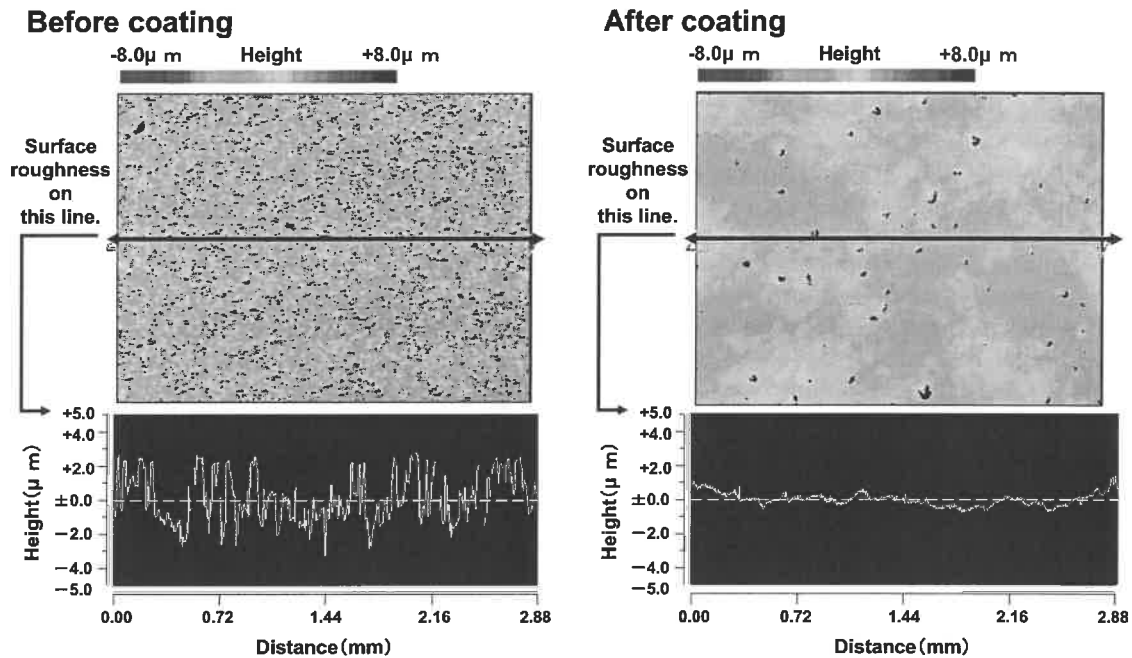


Figure 3. Measurement result of surface roughness before and after coatings with $N_u=2$ and $N_t=2$.

COATING EXPEREMENTS AND VISUAL EVALUATION OF TEXTURED METAL SHEET

We first textured the surfaces of four metal sheets with 400, 800, 1200 grit of sandpapers and a nail. We then coated the four sheets and measured their surfaces. We executed under and top coatings both twice, which was the best result obtained in the previous section, to the four sheets.

Table 2 shows the textured metal sheet surface roughness before and after coating. Sq means root mean square value, and Sz is sum of the largest peak height value and the largest pit depth value within a definition area. From the result, the similar qualities to those in the previous experiment were achieved in all four sheets. In addition, for observing adhesion condition of coatings on the metal sheet surfaces, we reinforced the coated metal sheet in resin, cut it, and then measured the cross section by field emission scanning electron microscope. Figure 4 and 5 show the adhesion conditions of coatings on the metal sheets textured by the 400 grits of sandpaper and by the nail respectively. The film thicknesses of two types of textured metal sheets were approximately 13μm and 12μm respectively.

Table 2. Textured metal sheet surface roughness before and after coating.

		No texture treatment	# 400	# 800	# 1200	Nail
Before coating	Sq	1.28	1.19	0.91	0.89	1.78
	Sz	15.26	29.69	11.94	10.39	46.54
After coating	Sq	0.44	0.59	0.53	0.48	0.82
	Sz	3.94	4.87	7.43	4.08	7.26

(Unit:μ m)

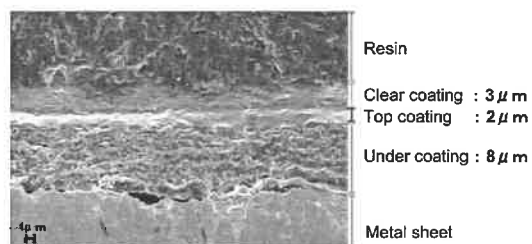


Figure 4. Adhesion condition of coatings on the metal sheet textured by the 400 grits of sandpaper.

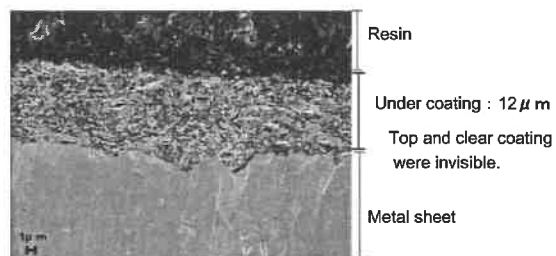


Figure 5. Adhesion condition of coatings on the metal sheet textured by the nail.

COATING EXPEREMENTS AND ADHESION EVALUATION OF TEXTURED METAL SHEET

We first created three patterns of textured metal sheet surfaces; single, 90° double, and 45° double with six texturing level, by using 80, 150, 220, 500, 800, and 1200 grits of sandpapers. And we executed under and top coatings both twice to these metal sheets, and measured the coated surfaces with a scanning white light interferometer. We next evaluated adhesion of metal sheet coating film by cross cut test [2]. Figure 6 shows the experimental method of cross cut test. And figure 7 shows the direction of adhesive tape to be peeled off. And we executed cross cut test to the nineteen coated metal sheets, including a non-textured sheet. And we then scored each of sheets from 0 to 5 according to the grading rules of cross cut test. Table 3 shows the metal sheet surfaces roughness and results of cross cut test evaluation. And figure 8 shows the examples of the metal sheet surfaces after cross cut test. As for the sheets textured by smooth sandpapers with 220 to 1200 grits, good scores (e.g. grading 0 or 1) were obtained independently on texturing directions and only minute chip off were observed at the intersection area of cutting (figure 8 (a), (b)). However, as for the sheets textured by rough sandpapers with 80 or 150 grits, relatively poor results (e.g. grading 2 or 3) were contained and relatively large chip off were observed along the single cut (figure 8 (c)) and in the rectangle surrounded by the four cuts (figure 8 (d)).

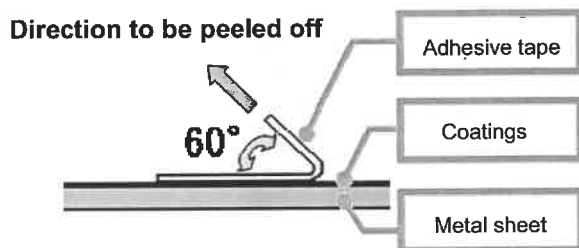


Figure 6. Experimental method of cross cut test.

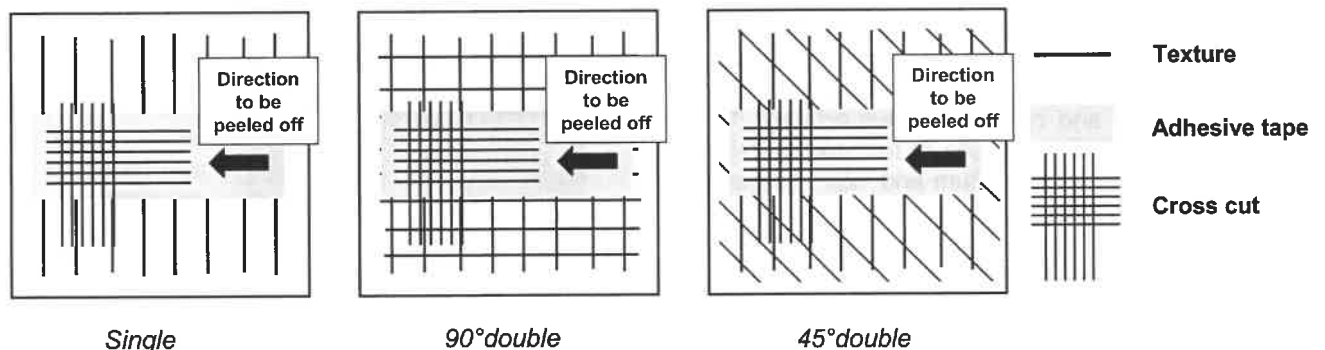


Figure 7. Direction of adhesive tape to be peeled off.

Table 3. Metal sheet surface roughness and result of cross cut test evaluation.

	Sq (Unit : μm)		Cross cut test Grading of 3 sheets (sheet 1, sheet 2, sheet 3)
	Before coating	After coating	
No texture treatment	1.169	0.422	(0, 1, 1)
#80 Straight	2.382	0.679	(0, 2, 3)
#80 Orthogonal	1.342	0.477	(0, 1, 2)
#80 45°Cross	1.220	0.553	(0, 1, 1)
#150 Straight	1.403	0.475	(0, 1, 2)
#150 Orthogonal	1.310	0.434	(1, 1, 2)
#150 45°Cross	1.109	0.313	(0, 1, 1)
#220 Straight	1.215	0.505	(0, 0, 1)
#220 Orthogonal	1.050	0.396	(0, 1, 1)
#220 45°Cross	0.944	0.454	(0, 1, 1)
#500 Straight	10.56	0.307	(0, 0, 0)
#500 Orthogonal	0.892	0.473	(0, 0, 0)
#500 45°Cross	0.798	0.599	(0, 0, 0)
#800 Straight	0.961	0.409	(0, 0, 0)
#800 Orthogonal	0.748	0.435	(0, 0, 0)
#800 45°Cross	0.748	0.318	(0, 0, 1)
#1200 Straight	1.075	0.515	(0, 1, 1)
#1200 Orthogonal	0.681	0.456	(0, 0, 0)
#1200 45°Cross	0.687	0.363	(0, 0, 1)

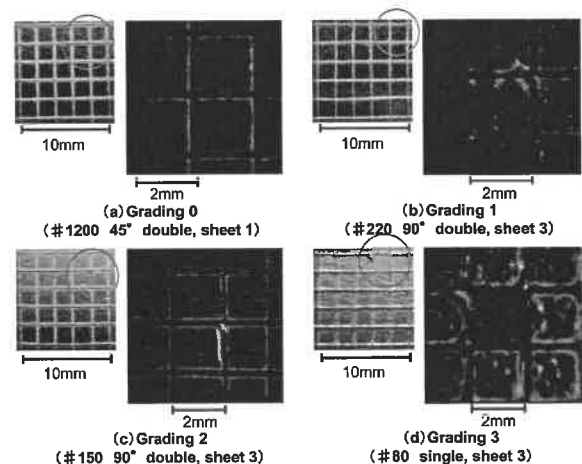


Figure 8. Examples of the metal sheet surfaces after cross cut test.

DIFFERENCE OF METAL SHEET SURFACE ROUGHNESS AND ADHESION BY THE DIFFERENT TIMES OF CLEAR COATING

To verify a relation between clear coating times and surface roughness and adhesion, we prepared the worst adhesion result of metal sheets, which was obtained in the previous section. This worst example was generated by texturing along single direction with 80 grits of sandpaper, and by performing under and top coating both twice. We then performed clear coating multiple times from 0 to 4 to this example, and evaluated adhesion by cross cut test. Figure 9 shows the surfaces of metal sheets after performing clear coatings 0 and 3 times. And table 4 shows the surface roughness and results of cross cut test after performing clear coatings multiple times. The surface roughness was pretty smooth except the one with no clear coating. From these results we found that clear coatings make the remaining roughness after under and top coatings smoother. For further adhesion, as for the metal sheet coating films without and after 1 time coatings, enough adhesions were not obtained. And as for the one after 2 times coating, only small film clippings were observed. And as for the ones after 3 and 4 times coatings, satisfactorily enough adhesions were obtained and chippings were not observed at all.

CONCLUSIONS AND FUTURE WORKS

In this work, we developed a trial coating robot and evaluated a relation between coating characteristics and surface textures. Our experiments were all performed under the condition of air pressure 0.3MPa, coating speed 50mm/s, under coating injection rate 4.5g/min, top and clear coating 3.0g/min. Under this condition, we found that the best number of coating operations were two for under and top coatings, and three for clear coatings. Moreover, we performed cross cut test and visually verified the relation between coating film adhesion and number of clear coatings or surface textures. In future work, we will quantitatively evaluate coating film adhesion by pull-off method.

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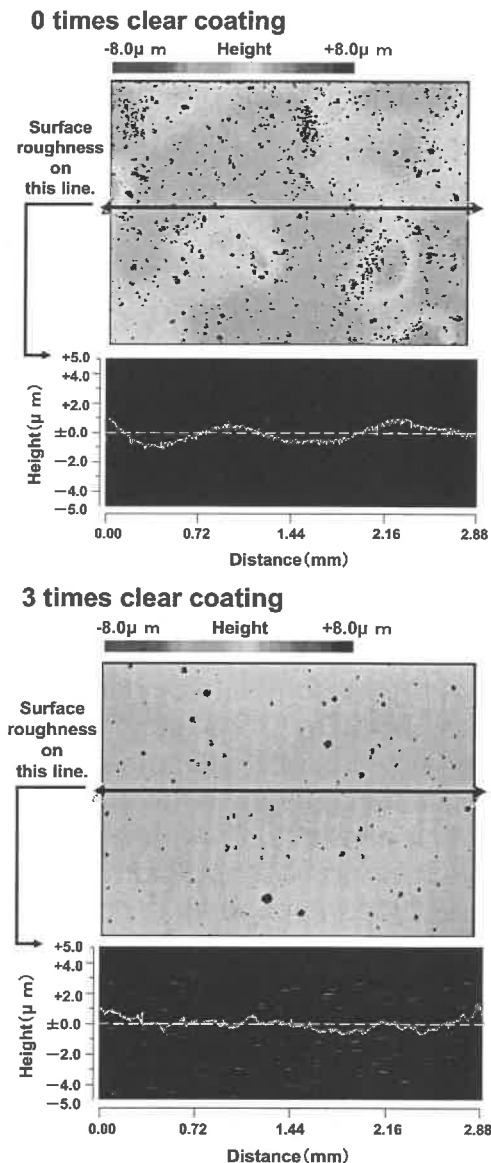


Figure 9. Surfaces of metal sheets after performing clear coatings 0 and 3 times.

Table 4. Surface roughness and results of cross cut test after performing clear coatings multiple times.

	After coating surface roughness	Cross cut test
	Sq (Unit : μm)	Grading of 3 sheets (sheet 1, sheet 2, sheet 3)
No clear coating	1.175	(6, 6, 6)
1 time clear coating	0.294	(4, 5, 6)
2 times clear coating	0.329	(0, 0, 1)
3 times clear coating	0.320	(0, 0, 0)
4 times clear coating	0.344	(0, 0, 0)

Application of Printed Electronics to SAW Devices on Mobile WiMAX/WiFi Systems

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1. Introduction

Because IEEE 802.16 had divided into two standards of WiMAX and mobile WiMAX, the situation that surrounded WiMAX was activated at a dash. IEEE 802.16e (mobility WiMAX) that IEEE is sure to superintend comes to be named, "Post 3.5G(Or, "3.9G" only)" before one is aware, and has come to exert the influence even on the area of the cellular phone that ITU (International Telecommunications Union) is sure to have superintended up to now.

2. WiMAX and mobile WiMAX

WiMAX and mobile WiMAX have surfaced as a potential threat as shown in Figure 1 for a past mobile telecommunications entrepreneur and IC supplier, etc. Figure 2 is a result of testing the field in the Yokohama city as for the power consumption of mobile WiMAX built into the portable terminal personal computer. Power consumption will become half in 2.2 hours after turning on the power supply, and power consumption will become half for the worst case in one hour. Figure 3 is Intel Wimax/WiFi Link 5150 module (Externals size: 50.95mm×30mm×3.3mm) built into the portable terminal personal computer. Figure 4 is an atom

transistor while developing to improve the low power consumption of the portable terminal personal computer. The atom transistor has achieved on/off by moving a little metallic atom in the insulator though a past transistor has achieved on/off by controlling the movement of the electron in the semiconductor. It on/off operates by supplying, and calling back the metallic atom from the gate electrode (upper part) to the source and drain electrode (blue lower part) side in the atom transistor as shown in Figure 4. The atom transistor can work because the insulator whose resistance is higher than the semiconductor is used for the parent metal, a small amount of metallic atom moves in it, and on/off is achieved by extremely low power consumption. However, FFT/IFFT can be done also with this atom transistor by using a cheap z transducer function of SAW (Surface Acoustic Wave). It has already proposed the OFDM/OFDMA system of Figure 6 where cheap z conversion of SAW was used. A chirp z transformation system of SAW adjusts the temperature coefficient of delay time to almost zero. SAW is a super-high-speed basically passive element. In the increase of the number of

channels of OFDM to 1024 or more, power consumption is extremely less than FFT/IFFT that consists of Digital FFT composed of the atom transistor.

3. Printed SAW device

The process of manufacture of the elasticity surface wave device is a method of transcribing the SAW pattern on a piezo-electric wafer by using [hotomasuku] by the photolithography technology, and it can be said a monotonous print in a wide meaning.

The screen process printing belongs to the having version print. An inkjet print is similar to the no version print method to form the SAW pattern directly on a piezo-electric wafer with the electron-beam printing here as the no version print. Figure 7 shows the process of manufactures of various SAW devices. In Figure 8, the inkjet no version print can form the aluminum thin film for which the

sound impedance matching is the best directly on a piezo-electric substrate as an elasticity surface wave device. The cost reduction of the amount of capital investment (SAW device manufacturing inspection equipment, high-grade clean room equipment, and utility equipment) by the sharp decrease and the turnaround time of the product can be greatly shortened by assuming the elasticity surface wave device product to be Disruptive innovation that not is so far.

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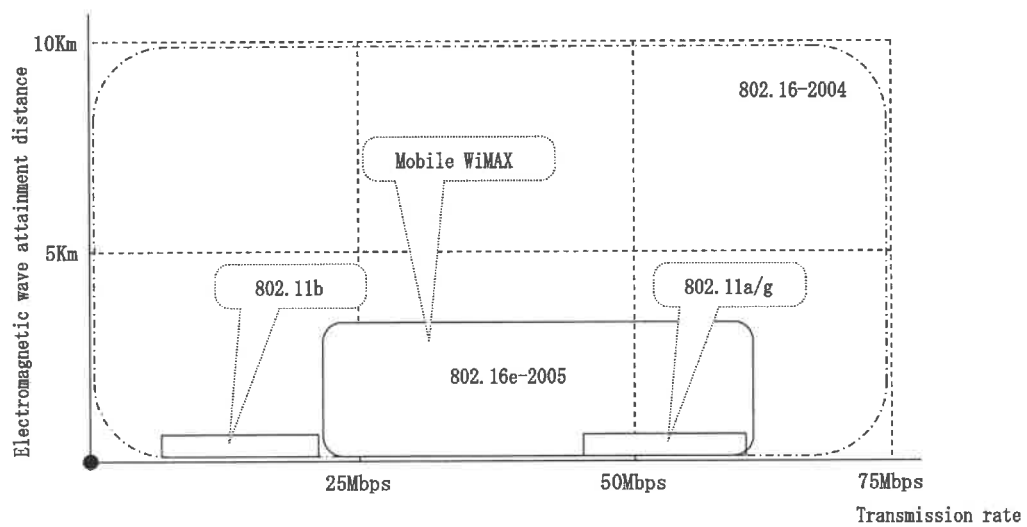


Fig.1 Radius of cell and the maximum transmission rate of WiMAX/Mobile WiMAX

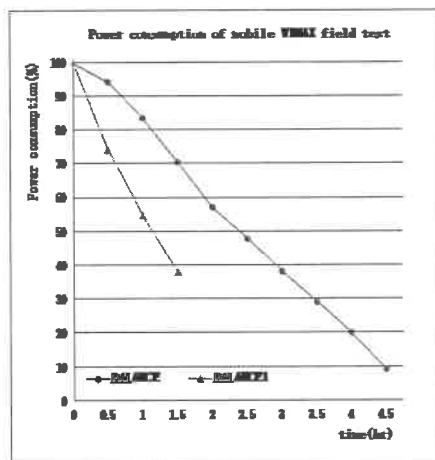


Fig.2 Power consumption of Mobile WiMAX in Yokohama city.

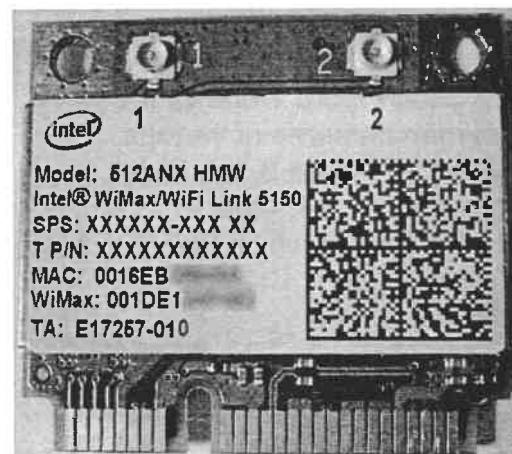


Fig.3 Intel WiMAX/WiFi Link 5150

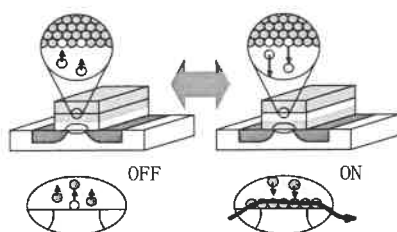


Fig.4 Atom Transistor

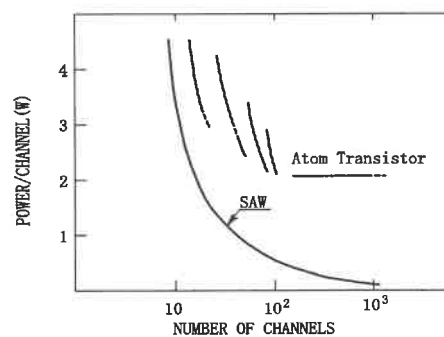


Fig.5 Power consumption SAW & Digital FET

Application of Precision Engineering in laser transverse pumping ^{133}Cs brain magnetometers

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1. INTRODUCTION

The first step of Table 1 shows various magnetic fields in the natural world, and an artificial magnetic field and the third step show the range of motion of the magnetic field weighing device in the second step. If the effect of Hanle of happening when the pumping magnetometer on electron does the diffusion conduction while shining and doing the precession movement is used, the sensitivity of 10^{-16}T can be obtained. Figure 1 and Figure 2 show the principle of transverse optical pumping. [1] Only slight element H_0 (Permanent magnet) of the direction of z remains about 0 when sweeping it so that the magnetic field element of the direction of x may pass 0 when a horizontal pumping to enter the circularly polarized light from the semiconductor of luminescence on the side laser into direct-current magnetic field H_0 vertically is done and the transverse magnetization is detected. In this paper, the optical pumping magnetometer researches the practical use of the diagnosis of the bodily function of the brain magnetism.

2. ^{133}Cs brain magnetometer

Figure 3 shows the magnetometer of ^{133}Cs developed brain. ^{133}Cs gas absorption cell that is the sensor of this brain magnetometer is put in a permanent magnet.

There is a transverse pumping to make the pumping light a circularly polarized light as a

method of greatly improving the sensitivity of the optical pumping magnetometer and to enter into the direct-current magnetic field vertically. As for the magnetometer of the brain, the one that operates at the same temperature as the temperature of the living body is desirable. 10ms or more is necessary to monitor the neuron activity the transverse relaxation time of the laser pumping of ^{133}Cs atomic vapor. The design of the system that measures the brain magnetism at 100fT level is as follows.

Figure 4 heats the operating temperature of the gas absorption cell of which the easing gas is neon to the Cs atom to 28°C – 46°C , passes the gas absorption cell the VCSEL (Vertical Cavity Surface Emitting Laser) light of 852.1 nm in luminescence wave length modulated in the gas absorption cell by the Raman transition frequency, and shows the relation between the sealing pressure and the easing time of the neon when Laser pumping is done. The easing time becomes the lowest at the time of a temperature it near 28.4°C in melting point of the Cs atom 28°C .

Figure 5 is a luminescence spectrum of VCSEL. Even if the temperature rises, the extension of the energy distribution of the electron is suppressed the effect of a thermal extension by the quantization and can lower the oscillation threshold to the quantum-dot structure of the VCSEL without causing the decrease in the gain to the utmost limit.

The photocount distribution that was wider than the Poisson distribution becomes a subPoisson distribution with only the half in the state of coherence by the negative feedback of the semiconductor laser before putting the negative feedback on the semiconductor laser. Figure 6 puts the negative feedback on the semiconductor laser and is a result of having precisely measured the noise level of photocurrent directly after the detecting phase as for the output light. A usual excitation method is asymptotic in the standard quantum limit of the coherent state. If the pump swing is suppressed with negative feedback loop, the standard quantum limit can be exceeded as shown in Figure 7. [2]

It is an essential requirement to always throw the current into the magnetic shielding to generate the effect of shaking in the magnetic shielding. The magnetic substance material with high permeability is developed in succession, and supermalloy is $\mu_{\text{max}}=6,000,000$ recently. Moreover, it has been described that meta glass 2605M S2 is $\mu_{\text{max}}=500,000$ without doing magnetism shaking. The magnetic shaking is not the main requirement that composes this paper.

Figure 8 shows the effect of the ionization current that happens by the dendritic process of about 50,000 neurons in the synaptic transmission.

The brain wave corresponds to DC-40Hz of the lamor frequency signals. The analogue signal is processed with the lowpass filter by the FIR method of the sampling frequency 80Hz. In this lowpass filter, the signal of the frequency 43Hz or more has been suppressed to 60dB or more. As for the signal that passed this lowpass filter, the image is processed as image brain magnetic wave signal of the A/D conversion. The processing of the lamor frequency signal consists of two-step

type A-D converter of the pipelined architecture. The A-D conversion of high speed and high resolution is achieved. It is sampled by track/holding (T/H) circuit, and the input signal is maintained as pipeline T/H, and controlled as an analog signal voltage value in the timing control part. After the voltage value is converted into a digital value of m bit with the pre-A-D converter, AD is converted into the analogue signal again with the D-A converter. The difference with the analogue signal that delays the analogue voltage value maintained in this analogue signal and track/hold circuit by an analog DeLay line is requested. The rest error amplifier converts the voltage of this difference into a digital value of m bit with the main A-D converter. The output of these two A-D converters is matched and the A-D conversion value of the m+n bit is obtained. The signal from ADA and main A-D enters the error correction machine. The signal from the error correction machine becomes a digital output. An analog delay line is necessary so that it may make amends for the operation delay time of the D-A converter following pre A-D converter and it. The detected brain wave makes the frequency a horizontal axis, and shows strength of the brain magnetic field of the alpha wave and the beta wave.

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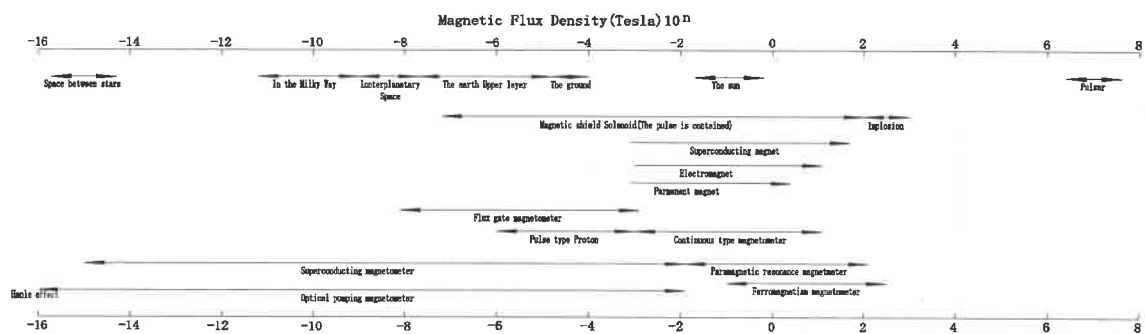


Table1. Range of magnetic field

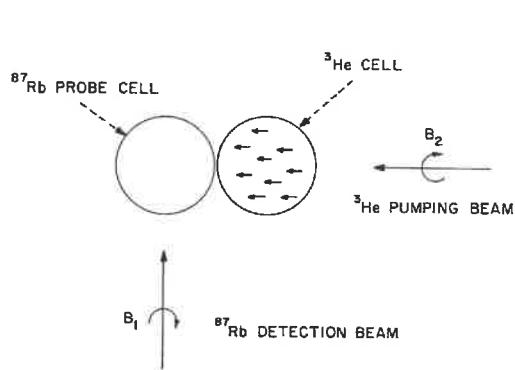


Fig.1 Transverse pumping principle chart

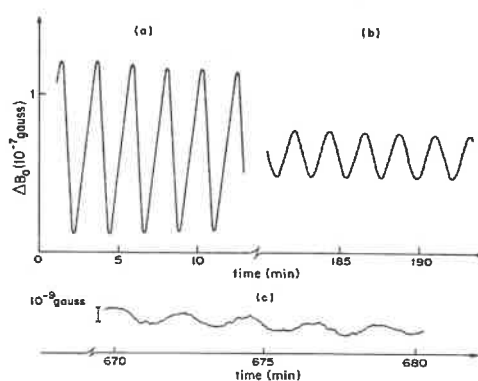


Fig.2 Result of a measurement

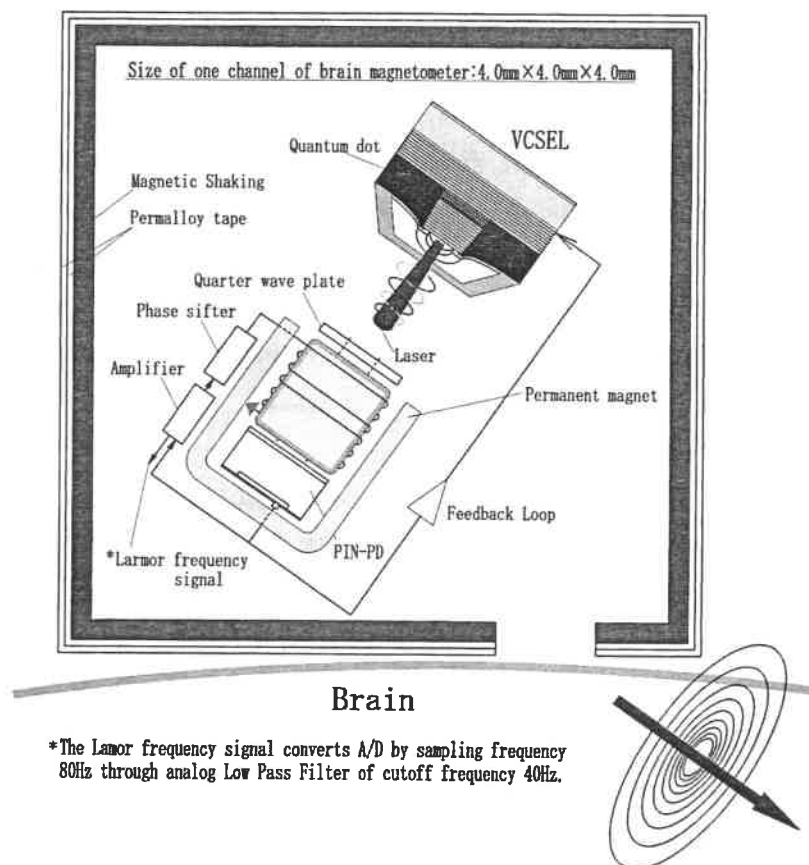


Fig.3 Developed ^{133}Cs brain magnetometer

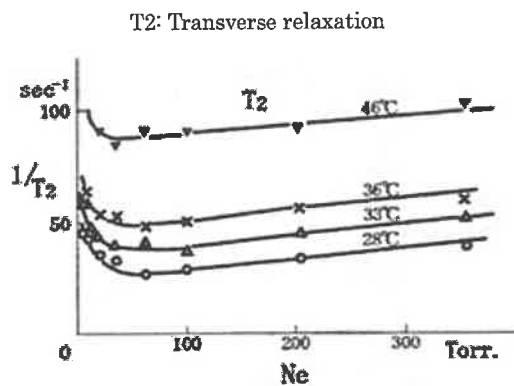


Fig. 4 Dependence of Ne buffer gas pressure of T_2

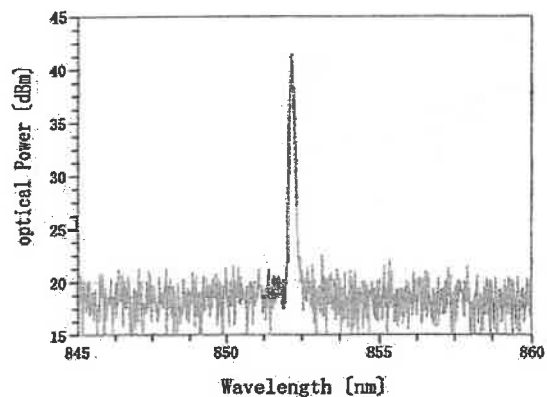


Fig.5 Luminescence spectrum of quantum dot VCSEL

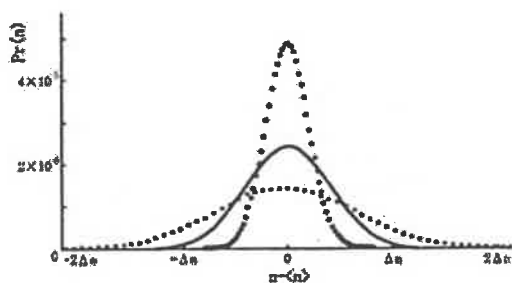


Fig.6 Photoelectron statistics for laser

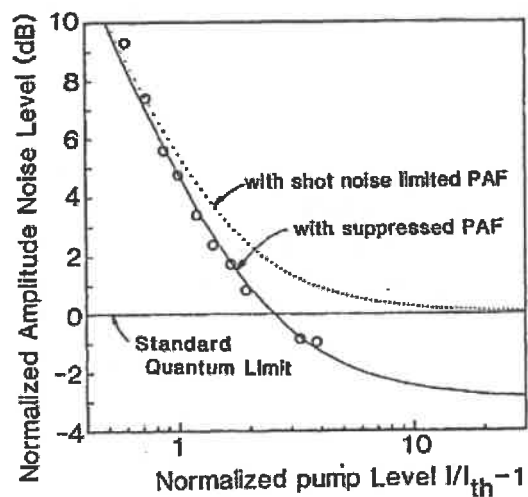


Fig.7 Squeezing of photon-number in feedback loop laser

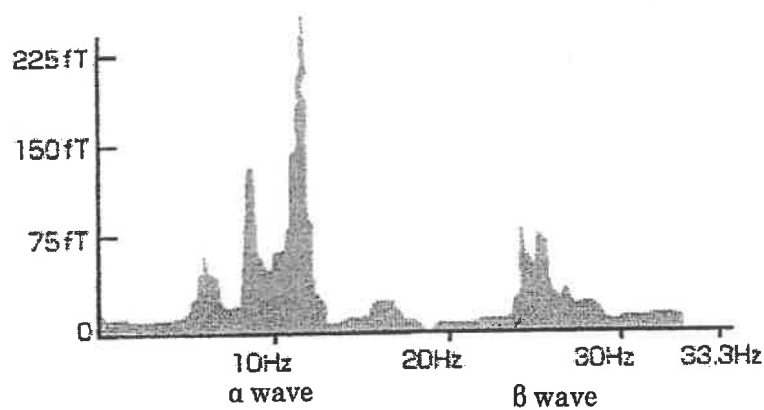


Fig.8 The measured brain waves

SIMULATION OF STYLUS CONTACT PATTERNS IN PROFILOMETRY

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INTRODUCTION

Stylus-based surface roughness metrology remains a popular and important practical technique for both industrial use and basic research, being complementary to optical methods. The nature of the contact at a small probe-tip is of clear concern to the fidelity of measurement, both directly in terms of potential distortion of functionally important data and for the inter-comparison of results from different instruments. Broadly similar concerns arise in some types of micro-tribology, but this paper concentrates on the former, while anticipating other more general applications. It is intuitively obvious that any surface features having dimensions, curvatures, *etc.* smaller than or even somewhat larger than those of the stylus tip will be distorted during measurement. Often regarded simply as a highly non-linear low-pass filter having (at best) poorly determined parameters, the probe interaction also has poorly understood effects on the measurement uncertainties, both normal to the surface (often called 'vertically') and in the surface plane ('horizontally' or 'laterally').

There are obvious, extreme challenges in attempting direct determinations of the complex interactions that occur between a profilometer stylus and a surface during a measurement of surface micro-topography. There is more hope that comparisons between real results and suitable numerical simulations might reveal indirect evidence about behaviour in the contact region. There is a long history (dating back to the obsolete E-system for generating measurement reference lines) of simulating the loci of the centres of circles, and latterly spheres, rolling on rough topography (see, for example, [1] for a brief review). It has been used primarily as an evaluation tool, but the concept also appears in attempts to compensate for stylus distortion by 'stylus deconvolution'. However, current models lack the sophistication to properly address comparisons or uncertainty analyses. They are, for example, purely

kinematic, assuming perfect guide axes, rigid probes and rigid surfaces. They are actually a class of morphological filter: specifically, the centre locus is dilatation operation [1].

THE NEW APPROACH

Previous stylus-surface interaction simulations have concentrated on how the apparent (actually measured) topography is modified by the stylus contact from what truly exists. There appears to have been very little research using such simulation methods to examine the effects on the stylus tip itself. This has been viewed as a lower priority question, but also it only makes sense to study it as a 3D simulation and so it has awaited the arrival of sufficient low-cost computing power. It is timely, therefore, to give it more attention. For instance, it offers potential to help build simple empirical models that might cast light onto questions of lateral uncertainty in stylus measurement, suggest ways to improve comparisons between different instruments, or lead to methods for self-diagnosing stylus wear.

The current work builds upon a previously reported 3D stylus simulation [2] that introduced a statistical summary of the points upon the stylus surface at which actual contact occurs. It allowed use of both simulated non-ideally-shaped tips and, particularly, the use of high-resolution data from direct measurements of real styli [3]. Such extra data becomes important because the contact is rarely directly along the (metrologically assumed) central axis of the stylus system.

The new simulation system introduces several refinements to this modelling approach. Firstly, it is clear that many simulations with subtle differences will be necessary to generate a statistically valid summary that could be compared to real instrument behaviour in order to infer the accuracy of the models. A highly modularized software suite is used, with bespoke parts implemented in MATLAB®. Figure 1 illustrates the scheme. Extensive use

is made of libraries of surface maps: in addition to simulated and measured 'original surfaces' every 'modified surface' processed here is also entered, so giving a complete record and also enabling study of multiple stylus actions. There is a similar library of 'styli'. Every simulation runs by loading an appropriate pair from the libraries, processing them by a core operation and returning modified versions and statistical information. Higher processing of surface information (for parameterizations, comparisons, etc.) is performed by a commercial package (in our case, SPIP®), which acts as an independent standard. This approach both maximizes capabilities for inter-comparisons from different sources and minimizes the size and complexity of the bespoke core routines, thereby easing the task of verifying them. For the present, we are content to work with small maps, up to around 128 points square, to limit computational costs.

The underlying starting point for modelling probe contact remains essentially the kinematic model used previously: an algorithm is published in [1]. The physical concept is that the central z-axis of a stylus map is positioned in turn above each (x, y) position on a surface map and then lowered along the z-axis until first contact is detected. This contact may not be on the z-axis but its height is recorded as the modified surface point at (x, y): this corresponds to where a real instrument would report such a point. Computationally, this dilatation operation reduces to an array addition and search for a maximum.

The first enhancement is to record also the position(s) (note that there might be more than one contact, e.g., where a sphere enters a triangular groove) of the contact within the stylus array. This can be done compactly by simply building a 2D histogram of contact numbers at each point [2], but more valuable information is gained by maintaining a history of contacts that correlates with the points on the surface map. It is then possible to revisit the same data sets using a separate bespoke module to explore in detail, e.g., the lateral error distribution between actual and reported height positions on the modified surface.

The kinematic model is unrepresentative and over-prescriptive because, among others, (i) real surface data is subject to instrument noise and so fine differences associated with the very peaks will not repeat; and (ii) neither surfaces

nor styli are perfectly rigid and there will be elastic (possibly plastic) deformation at contacts. Furthermore, it might be over-sensitive to the resolution of the surface data supplied by a specific instrument; a concern for measurement standards. A new threshold feature has been added to the kinematic model to address such points. On determining the kinematic condition, a search is made for other points which come within a specified distance of the stylus. This has immediate, obvious application in exploring how potential contacts might be lost or gained if resolution is reduced. It can also identify how many 'contacts' might be totally within the noise-floor of an instrument (physically the routine is one-sided, but the dilation has already effectively treated the noise as one-sided). Thresholding can also provide a first step towards more sophisticated models for approximating stiff but non-rigid contact. By a type of relaxation method, iterative modifications to the threshold might illustrate how the stylus settles elastically onto the surface. Each change of threshold would now be governed, e.g., by a force-balance using localized Hertzian contacts.

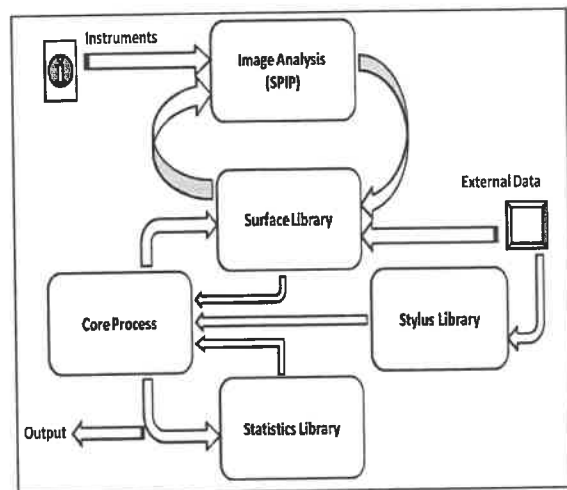


FIGURE 1. Scheme for the software suite.

ILLUSTRATIVE RESULTS

The basic simulation suite is now established for use in extensive numerical experiments. Here, some early results are provided as a first demonstration of its capacity and potential for further analyses. In attempting to make the concepts more immediately obvious, this paper shows only a physically unrepresentative simulated stylus tip shape, involving an ideal truncation (flat), and a relatively large threshold.

Verification

The challenge of adequately verifying proper operation is greatly eased by the use of small modules that are used sequentially. Running many trials of basic stylus shapes over simple surfaces, including single or clustered delta functions, rapidly identifies major bugs and increases confidence that no subtle errors remain. Just one example is given here, Figure 2. The pattern of a flat scanning across a noise-free sinusoidal prism is readily recognizable. It is also intuitively obvious that the contact pattern on the stylus has a low, uniform count (matching the flat regions on the modified surface) but there are higher values at the leading and trailing corner points (flank contact) and occasional multi-point contacts.

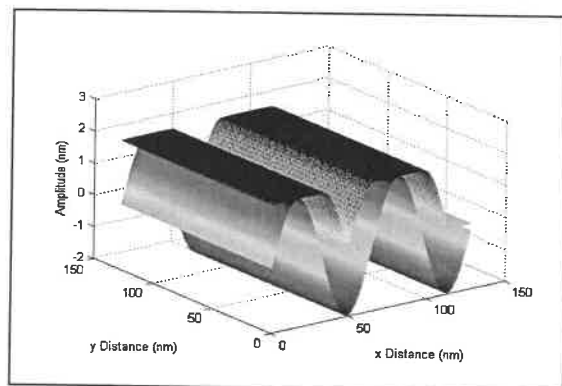


FIGURE 2. Verification – flat on sinusoid.

Kinematic and Threshold Models

Figure 3a shows a small map (128 by 128 points on a 0.2 μm sample grid) taken by atomic force microscopy from a typical ground steel surface. The fine detail has both sharply varying and plateau-like regions. For clarity of illustration, Figure 3b shows the severe effect of applying a $\sim 2 \mu\text{m}$ square flat stylus (11 by 11 points) under kinematic conditions. The serious distortion is no surprise, with high-points being converted to plateaus, but, even so, the general structure is preserved: e.g., peak-to-valley amplitude has reduced only from 758 nm to 746 nm.

A typical profilometer set up to measure finish ground surfaces might have an effective noise floor of the order of 10 nm to 20 nm peak. Its digitized resolution might typically be 1 nm to 10 nm. Alternatively, noise-related issues could be handled in terms of a fraction, say 0.5% to

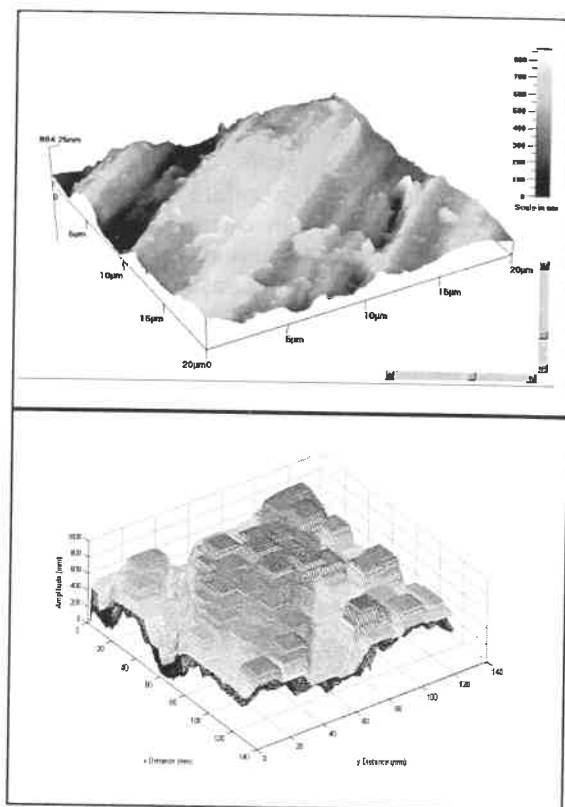


FIGURE 3. Modification of a surface detail by a 2 mm flat: (a) AFM image; (b) simulation

0.1%, of full range (maybe a few tens of micrometres, typically). Hence, thresholds from 5 nm to 50 nm (for deliberate over-emphasis) seem relevant for study here. The simplest form of thresholding is roughly equivalent to ideally pressing the probe into the surface. For the illustrations used here the modified surfaces are largely unchanged other than a slight vertical offset. Interest lies in how the shift interacts with additional surface features to alter the probe contact patterns.

As an elementary illustration, Figure 4 shows the effects on the position of contacts on the probe when adding a 50 nm threshold to the case shown in Figure 3b. For clarity, the maps show simply the presence or absence of contact at each point. Within the bounds of this flat probe, the actual area detected as 'in contact' is then highly analogous to the operation of the well-known bearing area. Note, though, that the pattern varies as the probe scans over the whole surface and more complex patterns will arise with more realistic probe shapes. Centring the probe at (61, 66) – using ordinates, not dimensions, for simulations – puts it on a steep

flank and kinematic contact is at one corner. Even 50 nm threshold causes only slight growth in the primary contact and the first signs of a second one nearby. By contrast, (61, 66) and (109, 120) are in plateau regions. They show large contact growth, dominated by a single region and occasional secondaries. In both cases, the kinematic contact is far from the centre of the region. Even at this small scale, the growing region develops a shape that correlates with the surface lay. In all three cases, the 'true' contact is significantly displaced laterally from the nominal line of action of the probe, a result also seen elsewhere [2] that has implications for measurement uncertainty of all topographic parameters except the simplest amplitude measures.

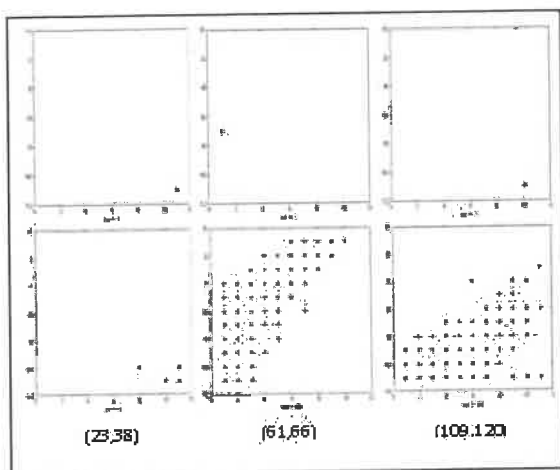


FIGURE 4. Positions of kinematic and 50 nm threshold contacts on a flat probe for three selected places in Figure 3b .

Although Figure 4 shows only the regions of contact, the intensity of contact at each point is of physical interest. One simple indication (not shown here) of this is to plot the probe maps in terms of how far each point on the probe would have encroached into the original surface to settle at the specified height relative to the kinematic condition. This gives, e.g., a crude indication of the stress intensities associated with each contact, and so of its likely practical significance in defining the reported surface height of a real instrument.

CONCLUDING COMMENTS

This paper has introduced a new implementation of a suite of tools for studying stylus-surface interaction in simulation. A consistent pattern of data flows and archiving around a highly

modular organization is advocated, allowing new features to be introduced in small, independently testable, modules. In this respect, the first features included and not previously reported concern the mapping of contact extent and intensity across the probe surface and the use of thresholding to investigate sensitivities to noise and instrument resolution. The concepts are illustrated for clarity by a highly artificial flat probe and large parameter values for which many effects can be understood intuitively, as well as being readily visible on small-scale plots. However, even with this unrepresentative approach, the thresholds reveal information about the surface geometry that may be of functional significance.

The system is now being refined to add further features while simultaneously being used with more representative probe models to investigate, for example, the statistical significance of contact distributions.

ACKNOWLEDGMENT

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COORDINATE METROLOGY BASED ON MAXIMUM CONFORMANCE TO TOLERANCES FOR CLOSED-LOOP FABRICATION

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INTRODUCTION

Compensation of systematic inherent errors of material-removal based processes is a practical approach for precise fabrication of the different scales of products from large to micro and nano levels. As a general concept, an on-line or intermittent inspection process of the semi-fabricated or unfinished surface is performed to complete a loop with the fabrication process. The results of inspection process are employed to develop compensating instruction for the next fabrication operations or phases. The essential requirements for estimation of deviation zone in the coordinate metrology process of a closed-loop system are discussed in this paper. Also, the importance of considering the desired tolerance zone in estimation of the optimum deviation zone is demonstrated. A best fit method is presented which complies with the tolerance requirements of the designed surface. The developed fitting methodology constructs a substitute geometry to minimize the residual deviations corresponding to the given tolerance zone and the needs of down-stream operations that use the results of the inspection process. It is shown how the developed objective function can be adopted for a case of closed-loop fabrication process, when the under-cut residual deviations of the manufactured part can be corrected by a down-stream operation.

In order to validate the proposed methodology, experiments are conducted. The results show a significant reduction of uncertainties in coordinate metrology of geometric surfaces. Implementation of this method directly results in increasing the accuracy of the entire tolerance evaluation process, and less uncertainty in fabrication of parts.

BACKGROUND

The classic fitting methods in coordinate metrology which fit substitute geometry to a set of the discrete points are performed by

minimizing a L_p -norm of the closest Euclidean distances as an error function, as follows:

$$L_p = \left[\frac{1}{n} \sum_i^n |\Delta(\mathbf{p}_i', \mathbf{G}'')|^p \right]^{1/p} \quad (1)$$

where n is number of the measured points, i is the index of the point $\mathbf{p}_i'$ measured from the actual surface, $\mathbf{G}'$, and $\Delta(\mathbf{p}_i', \mathbf{G}'')$ is the closest distance of this point from the substitute geometry, $\mathbf{G}''$. The standard definition of the tolerance zone is presented in ASME Y14.5 [1]. Numerous research attempts are focused on solving the stated problem to estimate geometric deviations. Least square ($p=2$) and minimum deviation zone functions ($p=\infty$) are typically utilized in the fitting process. The criterion in the least square function requires that the sum of square errors be minimized. The minimum deviation zone function has received a great deal of attention in recent years [2-3] because studies show that it yields a smaller zone value than that evaluated by using least squares fit [4-5]. In addition, it best conforms to the standard definition of the tolerance zone in ASME Y14.5 [1]. Furthermore, it numerically simulates the physical fitting process in traditional metrology. However, such an extreme fit has limitations in accuracy [6-7] and stability. It is also very sensitive to the number and the location of the measured points. Minimization of the deviation zone is a highly non-linear problem. Since the analytical derivatives of the objective function are not usually available, a direct search method is required to solve the problem and its success is very dependent on the initial conditions [8]. The least square best fit is the most likelihood estimation used to fit the substitute geometry onto a set of discrete data. Since all the measurement points contribute to the best-fit result, the substitute geometry is more stable and less sensitive to the local deviations, asperities and local surface effects. However, least square best fit is a statistical estimation rather than an exact solution and some

concerns always exist regarding interpretation of its fitting results.

MAXIMUM CONFORMANCE TO TOLERANCE

Minimizing the L_p -norm defined by Equation (1) does not consider the actual tolerance zone in estimating the geometric deviations, which results in improper corresponding points for the compensation process.

The developed fitting methodology constructs a substitute geometry to minimize the residual deviations corresponding to the given tolerance zone and the needs of down-stream operations that use the results of the inspection process. It is shown in this paper how the developed concept can be adopted for a case of closed-loop fabrication process, when the under-cut residual deviations of the manufactured part can be corrected by a down-stream operation.

Considering the requirements of the compensation phases in a closed-loop material-removal, a suitable fitting process is developed with the following properties:

- **Property 1.** eliminating the over-cut situation
- **Property 2.** minimizing the under-cut by minimizing the residual deviations

A new error function is defined for compensation purpose that complies with the required properties. The error function is used to describe an objective function which has a mini-max form. The resulting mini-max problem has no differentiable objective function and it needs to be solved by a direct search method. The developed objective function can be highly non-linear with discontinuities. In practice, the optimization may have an infeasible initial condition and stuck there. Also, it is likely to be stuck in a feasible pocket with a local minimum. In order to solve the second problem, a modified version of the objective function is presented, which temporarily relaxes the over-cut constraint to avoid straying too far from the feasible solutions. The first problem is a common problem in all of the highly nonlinear objective functions that are not differentiable, including the minimum deviation zone function. In this particular case, a method for iterative data capturing is utilized that can efficiently solve this problem [10]. The employed sampling method iteratively selects new points to be added to the fitting process. The new data points added in each iteration increase the energy level of the optimization process and boost the optimization to escape from the local minima [9].

In order to study the performance of the presented method variety of experiments are conducted and a simulation environment for the measurement process is developed. The developed methodology is a unique fitting function that constructs a substitute geometry to minimize the residual deviations corresponding to the given tolerance zone, and the needs of down-stream operations that use the inspection results. It complies with the basic requirements of the compensation process in the closed-loop fabrication and can be employed for precise fabrication of variety of products from large to micro and nano scales.

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Automatic Valve Inspection of Fuel Injection Nozzles with a Focus Variation Instrument

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INTRODUCTION

The leak-tightness of the valves for fuel injection nozzles is a crucial requirement to reduce fuel consumption and emissions. Due to the small dimensions and small tolerances of such nozzles, as well as the fact that the measurement is performed through a hole, the check for leaks is a demanding challenge. Here we present an optical measurement device based on the focus variation technology that measures the topography of the valves and automatically extracts significant parameters from this topography in order to judge the leak-tightness. In the following we first present the 3D measurement device used for the inspection and afterwards describe the inspection methodology.

3D MEASUREMENT WITH FOCUS VARIATION

The device used for the inspection is the optical 3D measurement device InfiniteFocus by Alicona (Fig. 1). Its measurement technique focus variation [1] [2] combines the small depth of focus of an optical system with vertical scanning to provide topographical and color information from the variation of focus. In contrast to tactile systems that have been previously used for many 3D measurement tasks the system is able to scan whole areas at once, a major advantage of several optical devices that have recently been developed [3]. Its vertical resolution depends on the chosen objective and can be as low as 10nm. The vertical scan range depends on the working distance of the objective and ranges from 3.2 mm to 22 mm. The x-y range is determined by the used objective and typically ranges from 0.14 x 0.1 mm to 5 x 4 mm for a single measurement. By using special algorithms and a motorized x-y stage the x-y range can be exceeded up to 100 x 100 mm and more. In contrast to many other optical techniques that are limited to coaxial illumination, the maximum measurable slope angle is not only dependent on the numerical aperture of the objective. However many different illumination sources (such as a ring light) are possible, which allow

the measurement of slope angles exceeding 80°. Since the technique is very flexible in terms of using light, most limitations when measuring surfaces with strongly varying reflection properties within the same field of view can be avoided.

In addition to the scanned height data, focus variation delivers a color image with full depth of field which is registered to the 3D points. This provides an optical color image which eases measurements as far as the identification and localization of measurement fields or distinctive surface features are concerned.

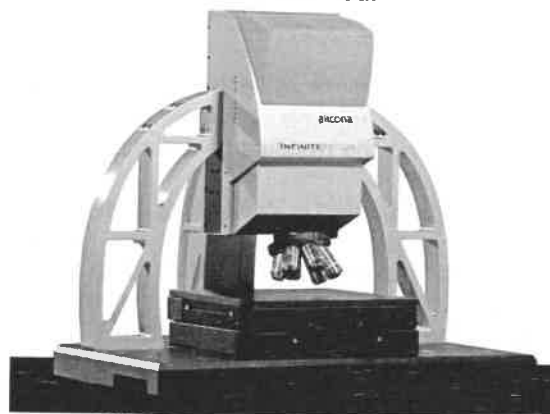


FIGURE 1. The InfiniteFocus system.

INSPECTION OF VALVES

The function of injection valves is schematically shown in Fig. 2. In Fig. 2a a sphere seals the valve so that no fuel will be injected. Afterwards (Fig. 2b) the sphere is lifted and fuel will be injected through the nozzle. The crucial part is to ensure that the sealing between the sphere and the valve is tight so that no fuel can leak through in step a.

In order to inspect whether the valve is tightly sealed a measurement software has been developed that automatically checks the quality of the valve. First a 3D measurement of the valve is performed and afterwards a series of parameters are automatically extracted to characterize whether the valve is good or bad. Some of these parameters are calculated for the complete cone-like valve, some of them are

calculated for the position where the sphere touches the valve.

Among the obtained parameters are values for

- roundness (RONa, RONq, RONT) [4]
- roughness (Sa, Sq, Sz, Spk, Svk, ...) [5]
- flatness deviation to cone (FLTt, ...)
- coaxiality and deviation to a sphere

The roundness is calculated according to the following procedure

1. High resolution measurement of the valve using focus variation
2. Determination of the position of a sphere falling into the valve
3. Automatic positioning of a cutting plane at the contact points of the sphere
4. Generation of the cutting profile
5. Waviness filtering of the cutting profile
6. Calculation of the parameters RONa, RONq, RONT.

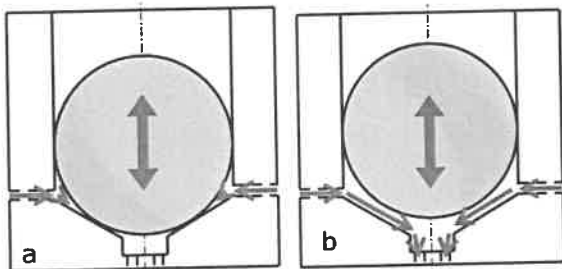


FIGURE 2. Schematic description of the two major states of the sphere within a fuel injector valve. a) Sealed state where no fuel (blue) is injected. b) State where fuel is injected.

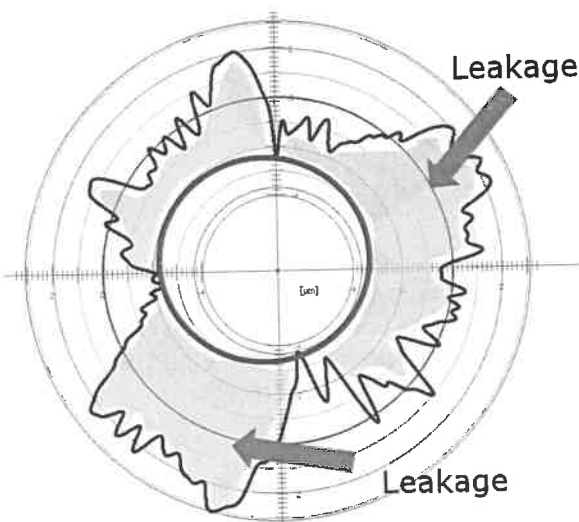


FIGURE 3. Roundness profile plot with schematic description of parts that produce leakage.

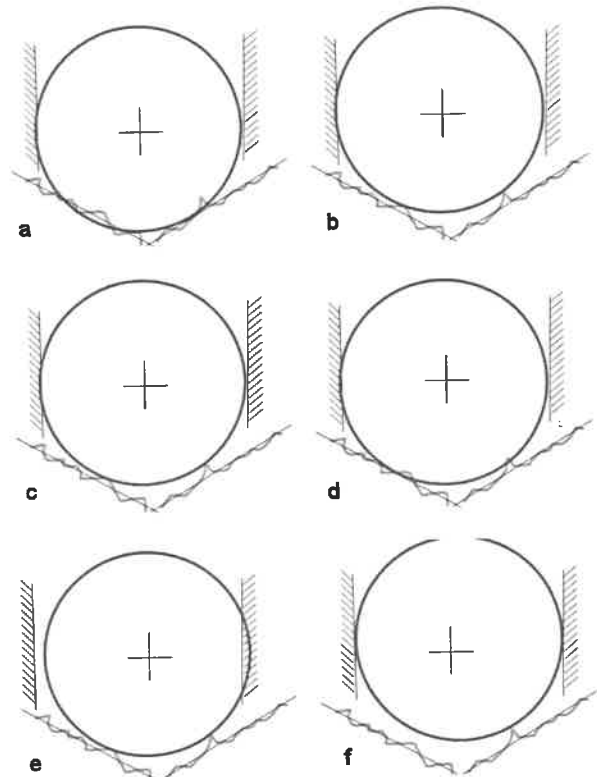


FIGURE 4. First row: Positioning of the sphere (a) by performing a least squares fit and (b) by letting the sphere fall down on the valve. Second row: (c) Positioning of the sphere by letting the sphere falling down without deformation and (d) with deformation of the valve. Third row: (e) Positioning of the sphere without guidance of the cylinder and (f) with guidance by the cylinder.

A schematic diagram of the roundness profile is provided in Fig. 3, showing the exaggerated deviations of the profile to a least-squares circle (outer red circle) fitted into the points.

The critical aspect during the roundness calculation is the calculation of the position where the sphere gets in contact with the valve. Therefore different algorithms have been implemented to calculate the position of the sphere after it has been positioned on the valve. One way to determine the sphere position is by fitting the sphere into the points of the valve cone that are closest to the sphere after letting it fall down (Fig. 4a). This method may not be physically correct since in reality the sphere does not move into the material of the valve but gets to lie on the cone. Therefore another algorithm has been implemented that searches the position of the sphere where the sphere ideally gets to lie on the cone (Fig. 4b). Since this method can be rather sensitive to small

irregularities (measuring artifacts, dust, etc) and may not be always physically correct (since the sphere will probably cut very small peaks), a method has been implemented that determines very small peaks, cuts these peaks, and lets the sphere fall onto the resulting surface (Fig. 4d). Final considerations have been made with respect to the way how and if the sphere is guided by the cylinder (third row in Fig. 4). In Fig. 4e the lowest position of the sphere is determined where the sphere may move in all three coordinates (x,y,z). However this this can lead to a result where the sphere would collide with the guiding cylinder. Therefore another method has been created that estimates the cylinder axis and searches those sphere position where the sphere touches the valve and the (x,y) center of the sphere lies on the axis of the estimated cylinder.

The flatness is measured for the whole measured valve (Fig. 5, top) after the conical form has been automatically removed.

In addition also the surface roughness (Sa, Sq, Sz...) of the valve is inspected according to latest ISO standards on area-based measurements [5]. For these measurements high resolution measurements of single positions are used (Fig. 5, bottom).

The software has been developed in a way that it can be easily adjusted to different valve types and different measurement parameters. Among the definable settings are for example the sphere diameter, the cone angle, the definition of the area on the cone for roughness measurement or the filter parameters for roundness calculation. Additionally the software is designed in a way that multiple valves can be positioned on a special sample holder and can be afterwards measured and analyzed in one run.

RESULTS

In order to evaluate the performance of the system various experiments have been performed. In one test the repeatability of the measurement parameters (roughness, roundness) have been evaluated, showing for example a repeatability of the roundness parameter RONA of $0.05\mu\text{m}$ at a mean value of $0.63\mu\text{m}$.

Additionally the parameters have been compared for good and defect parts as shown in Table 1. The calculated parameters allow a clear discrimination of the function of the part (good, bad) based on the extracted parameters. Good

parts had Sq values in the range of $0.15\mu\text{m}$ to $0.2\mu\text{m}$ whereas bad parts had Sq values that were between a range of $0.32\mu\text{m}$ to $1.02\mu\text{m}$. The RONq values showed a similar tendency with good parts having values up to $0.6\mu\text{m}$ and bad parts having values of about $1\mu\text{m}$ and more.

TABLE 1. Comparison of obtained parameters for good and bad parts. The bad parts have significantly higher Sq and RONq values.

Value 1	Sq [μm]	RONq [μm]
Good part 1	0.2	0.6
Good part 2	0.15	0.75
Good part 3	0.16	0.35
Bad part 1	0.32	1.14
Bad part 2	0.76	0.97
Bad part 3	1.02	1.7



FIGURE 5. 3D Measurement of valves from injection nozzles by focus variation. Top: measurement of the whole valve. Bottom: detailed measurement of one part with pseudo colors that represent the depth.

CONCLUSIONS

We have demonstrated a system for the automatic inspection of the leak-tightness of fuel injector valves. The system uses the focus variation technology which is able to measure also steep surfaces at high resolution. Due to

the special automation software it is possible to measure and analyze multiple valves in one run. The automatically extracted parameters have shown to provide a good discrimination between good and defect parts.

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METROLOGY OF BULLETS AND CARTRIDGE CASINGS

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INTRODUCTION

It is hypothesized that measuring the overall shape of a bullet and the three-dimensional geometry of cartridge casing features provides a fast and accurate way to match a bullet and/or casing to the weapon that fired it. The techniques that have been developed over the past 100 years are based on microscopic comparison of surface finish patterns. For a bullet, features in the rifling lands are inspected using a comparison microscope to look for matching patterns. For the casing, the shape and location of the firing pin impression, breech face features and extractor/ejector marks are compared.

High speed, non-contact measurements have been made of bullets and cartridge casings using the Polaris 3D measuring machine [1]. This machine measures the shape of a part by finding surface points in a spherical coordinate system; radius, latitude and longitude. It features a high-resolution chromatic aberration optical pen¹ to find the surface with a spot size and lateral resolution of 3.5 μm and vertical resolution of 3 nm. The optical pen is mounted on a linear axis and together they locate the radius of the part with respect to the machine origin. This radial axis is mounted on a rotary spindle that identifies the latitude angle. The part is mounted on a second rotary spindle that provides longitudinal position. The dynamic range of the optical pen along with real-time tracking control allows measurement of an imperfect shape with the part spinning at 100 rpm. Spherical, aspheric, cylindrical and non-rotationally symmetric surfaces can be measured. This unique geometry is ideal for measuring bullets and cartridge casing features and provides quantitative information, that is, dimensional metrology, to the firearms examiner.

¹Optical pens are available with different range, resolution and working distance specifications.

SCOPE

This work expands the range of measured parameters and adds statistical correlation and uncertainty to improve the capability of ballistic analysis. The solution is not simply a new sensor looking at the same features, but a new way to view the problem. A 45-cal firing pin impression is about 1.4 mm in diameter and 0.5 mm deep so photographs and microscope images can only capture a small region due to depth of field. Previous researchers have measured the striation patterns and developed suitable correlation techniques [2]. The Polaris 3D can record the location of a feature as well its detailed shape relative to any other feature on a bullet or cartridge casing – true 3D dimensional metrology.

PRELIMINARY RESULTS

To evaluate the capability of this machine to measure bullet features and firing pin impressions, firing tests were performed by the North Carolina State Bureau of Investigation and artifacts were provided to the Precision Engineering Center. Figure 1 shows a bullet (left) and the base of a cartridge case being measured by Polaris3D.

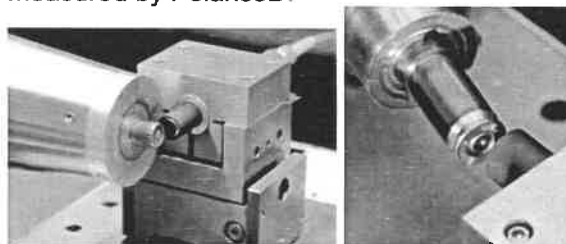


FIGURE 1. Bullet (left) and cartridge case (right) measurements on Polaris3D.

Bullets

Figure 2 shows the results of the bullet measurement from Figure 1. The overall shape differs by about 200 μm from a true cylinder. The color scheme of the plot indicates deviation from a fixed radius; in this case the best fit cylinder radius to all of the data, 5.68 mm. Figure 3 shows the measurement unwrapped from that cylinder and plotted in 3 dimensions as

radial deviation versus axial distance (z) and circumferential angle ($^{\circ}$). The maximum inscribed radius can be found by subtracting the smallest residual value from the cylinder radius.

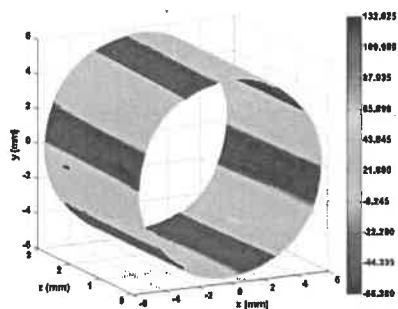


FIGURE 2. Bullet measurement. The dark bands are land impressions left by the barrel rifling.

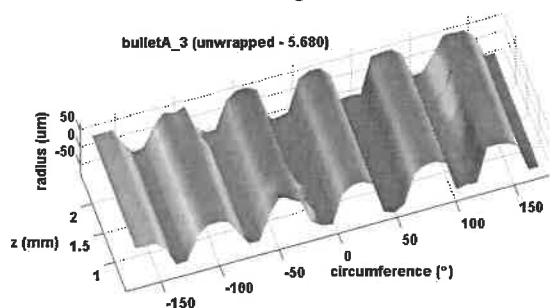


FIGURE 3. Unwrapped bullet features.

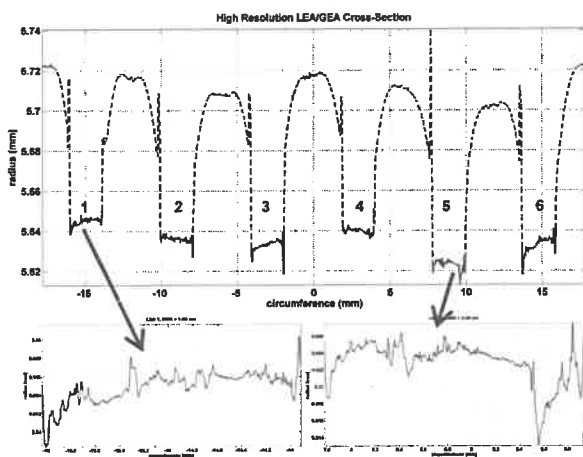


FIGURE 4. High resolution trace around the bullet circumference. The striation patterns from two of the rifling marks are shown.

Another way to look at the bullet is to take a single revolution of the part as illustrated in Figure 4. This data took about 30 seconds to collect and represents 33,000 points or about one point per micrometer of arc on the

circumference of the bullet.

Cartridge Case

The cartridge data from Figure 1 is shown plotted in relief in Figure 5. The impressions left by the firing pin and the "drag mark" made by the pin when the casing was ejected by recoil action are clearly visible. Figure 6 shows orthogonal cross-sections of the pin impression.

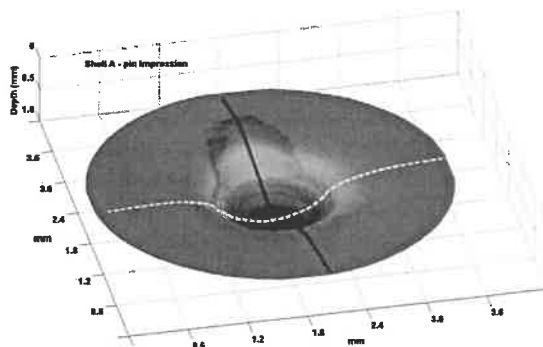


FIGURE 5. Impression left by the firing pin in the primer region of the cartridge case.

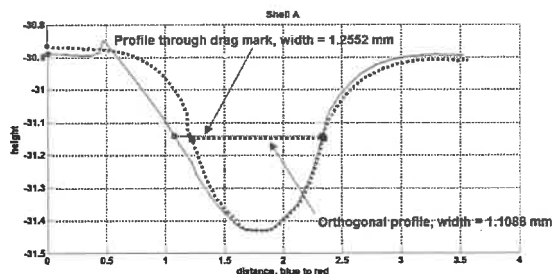


FIGURE 6. Orthogonal profiles through the firing pin impression.

CONCLUSION

Polaris3D rapidly measures both the traditional striation patterns seen by a firearms examiner in a microscope as well as geometric parameters that have yet to be explored as distinguishing characteristics. Expansion of the number and type of parameters with traceable and statistically reliable measurements will provide the examiner with a new tool for matching and identification.

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MULTI-PROBE ERROR SEPARATION APPLIED TO ROUNDNESS CIRCULAR FLATNESS AND ANGULARITY

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INTRODUCTION

Multi-probe error separation has particular utility for Reflective Electron Beam Lithography (REBL), a new technology being developed at KLA-Tencor under DARPA contract HR0011-07-9-0007. The REBL architecture features a continuously rotating platter of wafers which also slowly translates under a digital e-beam projector similar in concept to a digital light projector (DLP). Within a small ($100\text{ }\mu\text{m} \times 6\text{ }\mu\text{m}$) stationary field, it projects a dynamic, digital pattern that exposes patterns that are stationary relative to the moving wafers. The motion of the rotary-linear stage must be measured in six degrees of freedom in real-time to provide both feedback for controlling stage motion, including the magnetic spindle bearings, and feed forward for steering the e-beam. Ultimately the planar motion of each wafer must be tracked to the nanometer level with a portion allocated to stage metrology and error separation.

Another paper in this proceeding (Di Regolo, et al) describes the stage metrology system. Pertinent to this paper, three radial and four axial plane-mirror interferometers measure the combined motion and shape error of cylindrical and planar surfaces, respectively, on a rotating ring mirror. These measurements over determine the motion component, and the redundant information is used in the multi-probe method to identify the roundness and circular flatness of the mirror surfaces and to removed these shape errors from the measurements. In the same way, four read heads measuring tangential motion of the angular encoder ring allow the angularity to be identified and removed from the measurements.

For this application, there are several significant benefits of using the multi-probe method compared to other self-calibration methods such as Donaldson [1] and Estler [2], [3] reversals or the multi-step method [4].

- Since the stage operates under vacuum, additional mechanisms would be required to perform a reversal.

- Unlike an air bearing, for example, the motion of the magnetic bearing is controlled relative to the mirror surfaces rather than its bearing surfaces. Reversing the mirror would effectively reverse the spindle as well. Thus a second artifact and metrology system would be needed to perform the reversal.
- The multi-probe method can be performed periodically as part of the normal operation.

Many authors over the years have described the multi-probe method for separating artifact roundness from its motion; see for example [5], [6], [7], [8]. The concept is elegantly simple: arrange enough sensors to over-define the artifact's motion and then combine their signals to be insensitive to that motion. What remains is a scrambled signal of the artifact's shape, be it roundness, circular flatness or angularity. Inevitably the combined signal is missing some information about the shape. Known as harmonic suppression, certain spatial harmonics are poorly recovered, and the problem can be extreme for regularly spaced sensors. This issue will be examined later in the paper.

The descrambling process involves representing the shape as a parametric function of the rotation angle, typically a Fourier series. After expressing the function at all the sensor positions, they are algebraically combined to be consistent with the combined signals. That is to say, these phase-delayed functions and the corresponding measurements are projected into a subspace that is orthogonal to the artifact's motion. The parameters of the shape function may then be determined by data fitting within this subspace. To recover the motion, which is relevant for REBL, the shape is subtracted from the measurements, leaving only the motion component.

FORMULATION

We present a mathematical formulation of the multi-probe method that is more general than that found in most references. It readily extends to

higher numbers of sensors and shows that roundness, circular flatness and angularity may be expressed in the same form. We use the exponential form of the Fourier series to represent the shape estimate and it leads to a neat expression for the harmonic sensitivity, which expresses how well particular harmonics can be recovered. It is used both in selecting the angular positions of the sensors and in the descrambling process to recover the shape.

Beginning with FIGURE 1, four sensors are placed around a circle to measure radial, axial or tangential motion of the artifact at the indicated points. As the artifact rotates θ , each sensor m_i for $i = 1, 2, 3, 4$ measures a combination of its motion and its shape s , which Equation 1 represents for radial, axial and tangential directions, respectively.

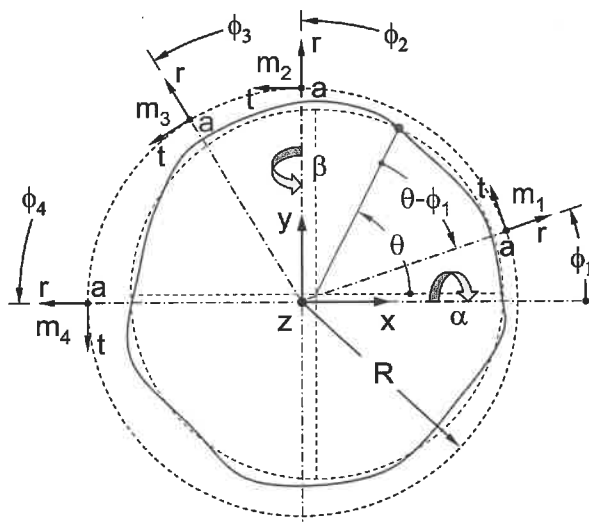


FIGURE 1. Positions of sensor measurements in radial, tangential and axial directions.

$$\begin{aligned} m_{r_i}(t) &\cong s_r(\theta(t) - \phi_i) + [x(t) \ y(t) \ \delta R(t)] \cdot \mathbf{a}_i \\ m_{a_i}(t) &\cong s_a(\theta(t) - \phi_i) + [-R \beta(t) \ R \alpha(t) \ z(t)] \cdot \mathbf{a}_i \\ m_{t_i}(t) &\cong s_t(\theta(t) - \phi_i) + \left[\frac{y(t)}{R} \ -\frac{x(t)}{R} \ \theta(t) \right] \cdot \mathbf{a}_i \end{aligned} \quad (1)$$

where

$$\mathbf{A} = \begin{bmatrix} \cos(\phi_1) & \cos(\phi_2) & \cos(\phi_3) & \cos(\phi_4) \\ \sin(\phi_1) & \sin(\phi_2) & \sin(\phi_3) & \sin(\phi_4) \\ 1 & 1 & 1 & 1 \end{bmatrix}$$

$[\mathbf{a}_1 \mid \mathbf{a}_2 \mid \mathbf{a}_3 \mid \mathbf{a}_4]$

We can see that artifact motion defined in the x-y-z coordinate system is transformed to the particular measurement direction. However, a number of small error terms have been neglected. Take for example the radial sensor at position 2,

which is nominally perpendicular to and first-order insensitive to x-axis motion. Unaccounted errors include: misalignment of the sensor, the nominal curvature and local slope error of the artifact, and the slightly different track that each sensor sees. Similar errors exist for the other sensor directions.

Applying the multi-probe concept, the first-order effect of artifact motion is eliminated as a linear combination of Equation 1 for all the sensors of a given type (e.g., radial). A coefficient vector $\mathbf{c}$ that satisfies Equation 2 will reduce the linear combination to that expressed in Equation 3.

$$\begin{aligned} \mathbf{A} \cdot \mathbf{c} &= \mathbf{0} \\ \mathbf{c}^T \cdot \mathbf{c} &= 1 \end{aligned} \quad (2)$$

Notice that the subscripts r , a , and t have been dropped since (3) applies to all three. Vector $\mathbf{c}$ may be solved using straightforward matrix inversion and subsequent scaling to unit length.

$$\begin{aligned} m_c(t) &\equiv \mathbf{m}(t) \cdot \mathbf{c} \cong s(\theta(t) - \phi) \cdot \mathbf{c} \\ \text{where} \end{aligned} \quad (3)$$

$$\begin{aligned} \mathbf{m}(t) &= [m_1(t) \ m_2(t) \ m_3(t) \ m_4(t)] \\ s(\theta - \phi) &= [s(\theta - \phi_1) \ s(\theta - \phi_2) \ s(\theta - \phi_3) \ s(\theta - \phi_4)] \end{aligned}$$

Taking the Fourier series of (3) over an integer number of revolutions results in Equation 4 and an expression for the harmonic sensitivity κ . So long as the magnitude of κ does not become too small, each Fourier coefficient S_k can be determined from the measurements M_{ck} using (4), since each harmonic is independent of the others. The inverse Fourier transform recovers the shape estimate of the artifact.

$$\sum_{k=-n}^n M_{ck} e^{ik\theta} \cong \sum_{k=-n}^n \kappa(\mathbf{c}, \phi, k) \cdot S_k e^{ik\theta} \quad (4)$$

where

$$\kappa(\mathbf{c}, \phi, k) \equiv c_1 e^{-ik\phi_1} + c_2 e^{-ik\phi_2} + c_3 e^{-ik\phi_3} + c_4 e^{-ik\phi_4}$$

DISCUSSION

When evaluating κ for different k , you will find that $|\kappa| = 0$ for $k = -1, 0$ and 1 . These correspond to artifact motion and by design and by virtue of Equation 2, they are absent from Equation 4. It is good practice to test the data to confirm whether motion is in fact being eliminated. If not, then there is opportunity to use the data to improve knowledge of the sensors' angular positions and possibly mismatching gage factors.

Readers familiar with the three-probe (or three-point) method for roundness may be perplexed as

to why four sensors are presented here, and in fact, three are being used for REBL. An extra degree of freedom is being included, i.e., uniform radial expansion of the artifact, which might occur due to changing temperature or speed. All radial sensors see the same expansion, analogous to the other directions for pure axial or pure angular motion. Without this degree of freedom, any expansion over the span of data collection, although unlikely, would be folded into the shape estimate in some inappropriate way. When it is preferable to disregard radial expansion, the row of ones is removed from matrix **A**. If three sensors are used, then the last column is also removed. In addition, $|k| \neq 0$ for $k = 0$.

It is worthwhile considering redundancy greater than one, as in principle better error separation and reduced noise are possible. Matrix **A** would expand with additional columns to match the number of sensors, and vector **c** would have freedom to be optimized. We did not pursue higher redundancy for REBL to avoid a more complicated interferometer system.

If the angular spacing between sensors is regular, a repeating pattern will occur in the harmonic sensitivity about multiples of n , where $1/n$ is the largest increment that divides evenly into all the angular spaces. Those harmonic sensitivities intended to be zero, to eliminate motion, repeat themselves at higher spatial frequencies according to this pattern, as FIGURE 2 shows. In the first example, the angular spaces 90° , 20° , and 250° are divisible by 10° or $1/36$ revolution. Thus spatial frequencies 35, 37, 71, 73 ... are unobservable and the corresponding amplitudes cannot be recovered. In the second example, the angular spaces 60° , 30° , and 270° are divisible by 30° or $1/12$ revolution, leaving 11, 13, 23, 25, 35, 37 ... spatial frequencies unobservable. These spatial frequencies must be excluded from the shape recovery process to avoid generating spurious amplitudes from amplified noise.

DESIGN PROCESS

It should be clear from the previous example that selecting appropriate angular spaces between sensors is a key ingredient to achieving good error separation and is an important design issue in applying the method. Other important design considerations include: dimensional controls to ensure that unaccounted errors (mentioned earlier) are indeed negligible; favoring sensitive directions over less sensitive ones; whether to use one or more degrees of redundancy; and making reasonable simplifications when possible. For

example, the REBL #1 rotary stage has all but one set of radial and axial interferometers aligned to orthogonal X and Y axes. The best odd-angle interferometer was determined from a single-parameter search to be about 1° from a 45° diagonal. Quite good results for this configuration appear later in FIGURE 4 and FIGURE 5.

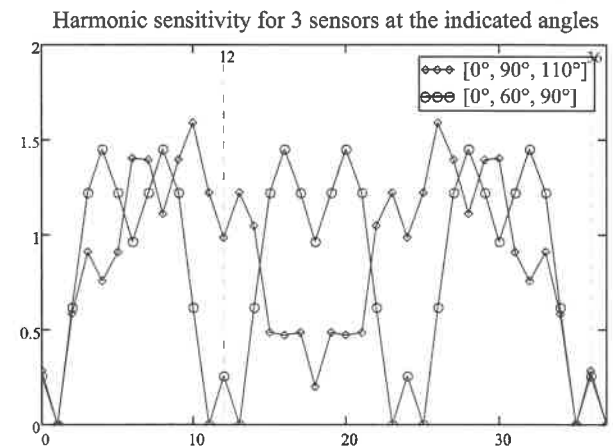


FIGURE 2. Examples of harmonic suppression.

The main focus here on will be setting up an optimization to determine the sensor angles. We develop a figure of merit based on the harmonic sensitivity κ over a range of spatial frequency. The temptation is to set a broad frequency range but inevitably this lowers the harmonic sensitivity to key lower frequencies and reduces the robustness to the sensors' angular position tolerances for higher frequencies. Ideally the range should be just enough to capture the shape of the artifact.

Equation 5 defines what we call the p-mean (after the p-norm or vector norm) of the harmonic sensitivity. It is like a signal to noise ratio that is to be maximized, and since noise is in the denominator, choosing $p = -2$ gives the signal over the total RMS noise included in the range.

$$m_p(\Phi) = \left\{ \frac{1}{n-1} \sum_{k=2}^n |\kappa(c(\Phi), \Phi, k)|^p \right\}^{\frac{1}{p}} \quad (5)$$

The nature of the optimization yields many local maxima, and the best peaks are both tall and broad so as to be robust to angle tolerances. So as a further enhancement, the figure of merit given by Equation 6 has breadth included by adding angular tolerance to the angle parameters with columns of **B** (shown here for three sensors).

There are various ways to implement such an optimization. We evaluate over a 2-D parameter space and report the maximum. The results are

visualized with a contour plot such as FIGURE 3 and with cross sections through candidate areas.

$$f(\phi) \equiv \frac{1}{7} \sum_{j=1}^7 m_p (\phi + \phi_{tot} \mathbf{B}^{(j)})$$

$$\mathbf{B} \equiv \begin{bmatrix} 0 & 1 & -1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 1 & -1 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 & -1 \end{bmatrix} \quad (6)$$

Figure of merit vs. probe angles

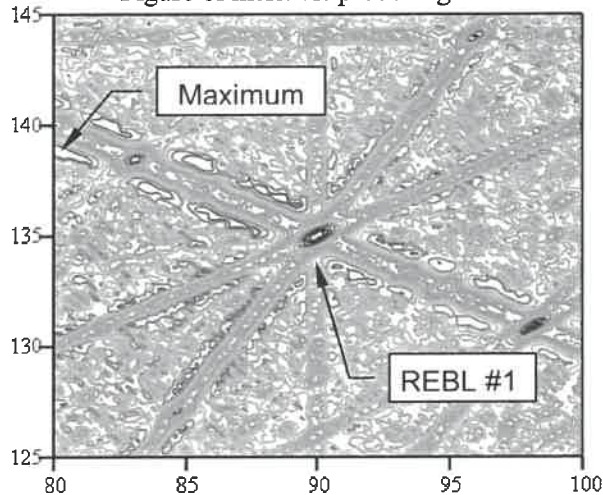


FIGURE 3. The maximum peak of 0.613 occurs at $\theta = [0^\circ, 80.4^\circ, 138.6^\circ]$. The REBL #1 peak of 0.516 occurs at $\theta = [0^\circ, 90^\circ, 133.8^\circ]$.

As seen in FIGURE 4, it is wise to plot the harmonic sensitivity over the frequency range of interest to confirm that none are too small, perhaps >0.1 . In addition, the p-mean should be plotted over a tolerance range for each angle.

Harmonic sensitivity vs. spatial frequency

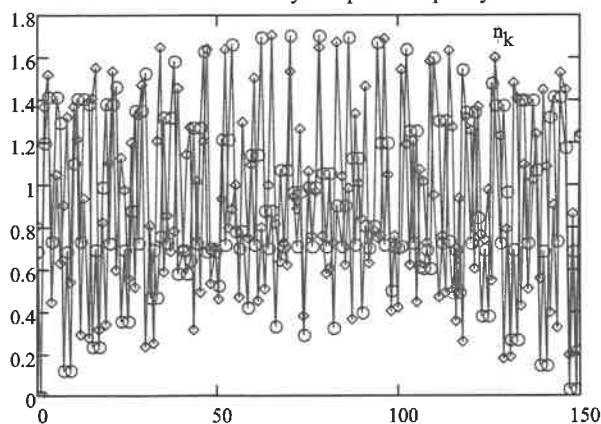


FIGURE 4. Plots of harmonic sensitivity for the maximum peak (red diamonds) and the REBL #1 peak (blue circles). Note, $n_k = 128$ was the spatial-frequency range used in the optimization.

EXPERIMENTAL RESULTS

Preliminary experimental results for the recovered shape of the ring mirror appear in FIGURE 5.

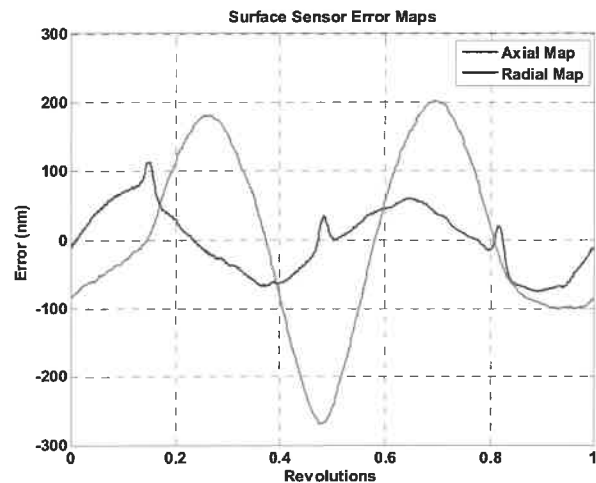


FIGURE 5. Axial and radial error maps of the ring mirror estimated for the REBL #1 rotary stage.

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PRECISE SORTING SYSTEM OF ROLLING ELEMENTS FOR BEARING

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INTRODUCTION

Recently, demands have been increasing for improvement motion and positioning accuracy of motion control devices, such as gear reducers and actuators, for factory automation facilities. As various bearings are used for essential parts of the devices, to improve the motion accuracy of the bearing is indispensable. However, the performance of commercial bearings is unsatisfactory due to the insufficient geometric accuracy of the parts. Therefore, to improve the motion and positioning accuracy of the motion control devices, it is necessary not only to improve the machining accuracy of the outer and inner race but also to minimize the shape error and mutual error of the rolling elements used for the bearing.

In this paper, a new sorting system for rolling elements is proposed using an originally developed angle sensors and the usefulness of the system is discussed.

PRINCIPLE AND STRUCTURE OF SORTING SYSTEM

A schematic view of the sorting system proposed for rolling element is shown in Figure 1. The dimensional difference in diameters between an objective rolling element and a reference one, both of them are placed on a flat plane, is measured by the inclination of a lever placed over the both elements. The expansion magnification of the dimensional difference can be enlarged by shortening the interval between two rolling elements. Then, the system can be miniaturized. In addition, the angle sensor is comparatively cheap and realizes high resolution by applying the theory described later. The light from the laser diode passes the

polarization beam splitter and 1/4 wavelength plate and radiates a mirror placed on the lever. A two dimensional angle sensor can measure the inclination angle of reflected light from the mirror due to the dimensional difference and the taper angle of the roller at the same time. In addition, an autonomous proofreading is also possible by the measurement after the replacement of two rollers.

Figure 2 shows the example of the sequential ball sorting system. Balls to be measured are successively fed passing on a linear track. When a couple of balls come to the measuring point, the difference in diameter is measured with the angle sensor as the inclination angle of the lever that touches over the adjoining two balls. Effect of linearity error of track surface can be calibrated using a reversal method by changing the order of the two balls.

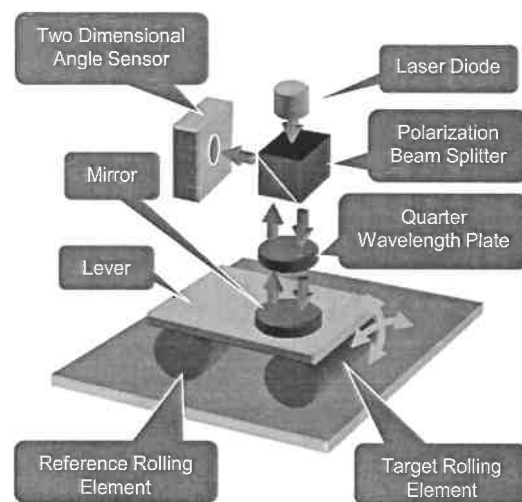


FIGURE 1. Roller sorting system

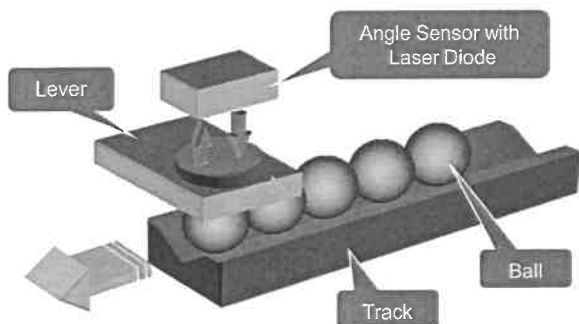


FIGURE 2. Sequential ball sorting system

Angle Sensor

On the interface where light is transmitted from the medium of high refractive index to that of low, at the incident angle around the critical angle, the reflectance varies remarkably by the change of a slight incident angle. A prototype angle sensor with high resolution was designed using the principle [1]. The composition of the angle sensor is shown in Figure 3. The light that is incident to the prism with the incident angle around the critical angle splits into reflected and transmitted lights, and the intensity of each light is detected by photo diodes A and B. By a differential amplification between the outputs of two photodiodes, the change in incident angle to the prism can be detected with high resolution as shown in Figure 4.

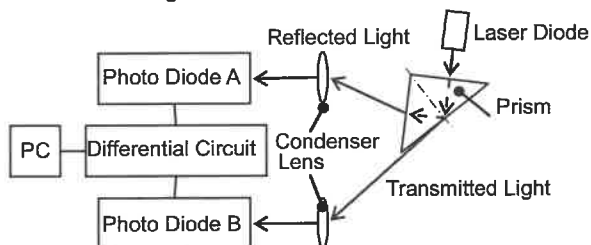


FIGURE 3. Composition of reflectivity type angle sensor

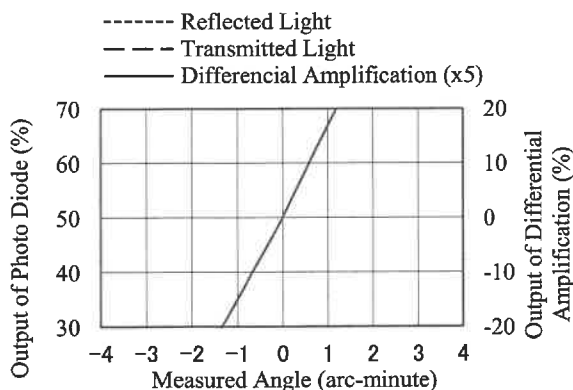


FIGURE 4. Angle sensor output in the vicinity of critical angle

Figure 5 shows the testing set-up of the performance of the angle sensor. The sensor was mounted on a rotary table and a laser source was so fixed near the table as the incident angle of the laser beam is around the critical angle of the prism. The table was rotated by a direct drive motor at the different angle at the step of 5.0 arc-second. Figure 6 shows the results of the testing. The sensor has sufficient linearity within the working distance of 50 arc-second. The noise level of the sensor was 0.1 arc-second as shown in Figure 6. Therefore, the resolution of the angle sensor is estimated to be 0.2 arc-second.

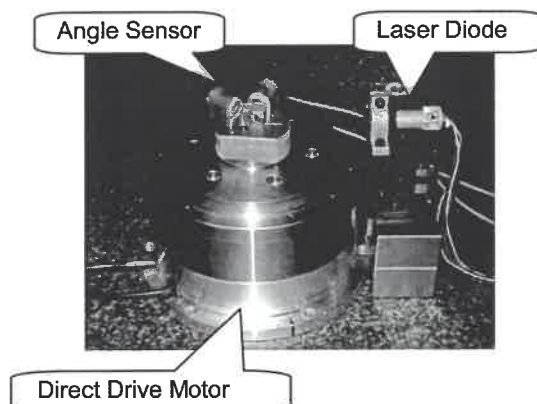


FIGURE 5. Step rotation test setting of angle sensor

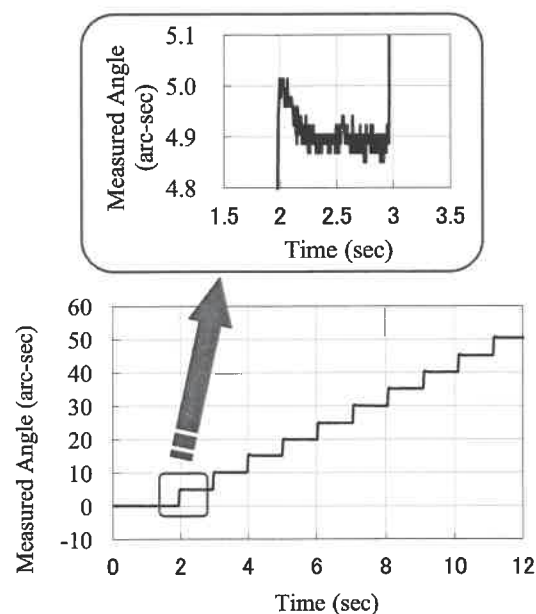


FIGURE 6. Angle sensor measurement result

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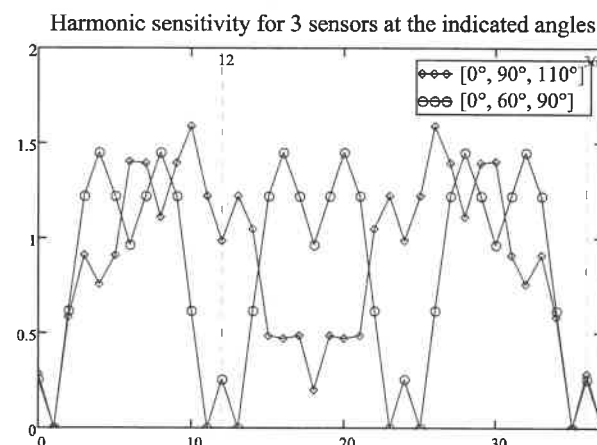


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The main focus here on will be setting up an optimization to determine the sensor angles. We develop a figure of merit based on the harmonic sensitivity κ over a range of spatial frequency. The temptation is to set a broad frequency range but inevitably this lowers the harmonic sensitivity to key lower frequencies and reduces the robustness to the sensors' angular position tolerances for higher frequencies. Ideally the range should be just enough to capture the shape of the artifact.

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There are various ways to implement such an optimization. We evaluate over a 2-D parameter space and report the maximum. The results are

visualized with a contour plot such as FIGURE 3 and with cross sections through candidate areas.

$$f(\phi) = \frac{1}{7} \sum_{j=1}^7 m_p (\phi + \phi_{tol} \mathbf{B}^{(j)})$$

$$\mathbf{B} = \begin{bmatrix} 0 & 1 & -1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 1 & -1 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 & -1 \end{bmatrix} \quad (6)$$

Figure of merit vs. probe angles

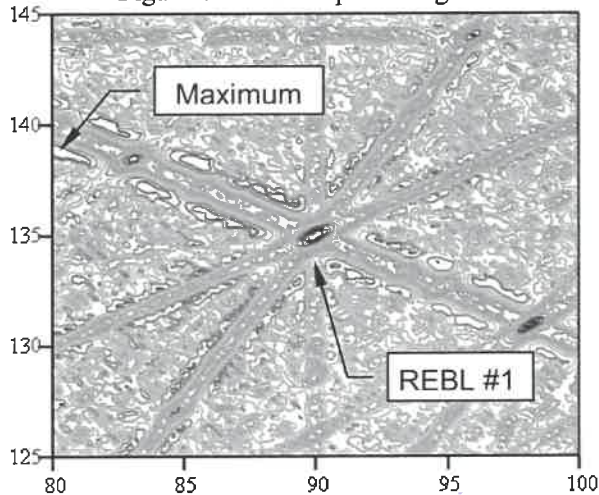


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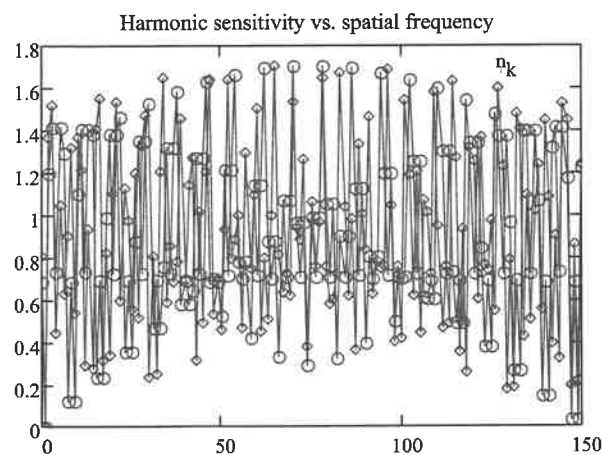


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Preliminary experimental results for the recovered shape of the ring mirror appear in FIGURE 5.

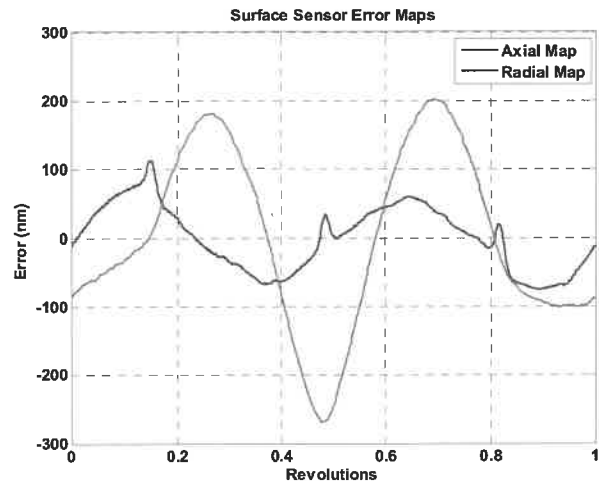


FIGURE 5. Axial and radial error maps of the ring mirror estimated for the REBL #1 rotary stage.

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ERROR MOTION ESTIMATION OF A HYDROSTATIC SPINDLE

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INSTRUCTIONS

The error motion of a machine tool spindle directly affects the surface errors of machined parts. The error motions of the spindle are not desired errors in the three linear direction motions and two rotating motions. Those are usually due to the imperfect of bearings, assembly errors, external force and unbalance of rotors. The error motions of the spindle have been needed to be decreased to desired goal of spindle's performance. The level of error motion is needed to be estimated during the design and assembly process of the spindle.

In this paper, the estimation method for the five degree of freedom (5 D.O.F) error motions of the spindle is suggested and applied to an hydrostatic spindle. To estimate the error motions of the spindle, waviness of shaft and bearings, external force model was considered. And, the estimation models are considering geometric relationship and force equilibrium of the five degree of the freedom. To calculate error motions of the spindle, not only imperfection of the shaft, bearings, such as rolling element bearing, hydrostatic bearing, and aerostatic bearing, but also driving elements such as worm, pulley, and direct driving motor systems, were considered.

FIVE D.O.F ERROR MOTION MODEL OF SPINDLE

To make the general motion model of spindle, not only location but also numbers of the bearings are considered to be defined as shown in Figure 1. The geometrical relationship is explained as equations (1) and (2). Equation (1) explain the displacement of shaft on the each bearing parts and equation (2) indicates the shaft in the point that users want to know.

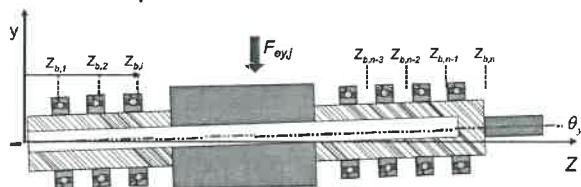


FIGURE 1. Geometrical models of a spindle

Where the subscript i means the orders of the bearing from the left and 1 means No. 1 bearing on the left side of the spindle. In here, θ_x and θ_y are tilt error motion rotating x or y axis, respectively.

$$\begin{aligned}\delta_{bxi} &= \delta_{bx1} + (z_{bi} - z_{b1}) \cdot \theta_y \\ \delta_{byi} &= \delta_{by1} + (z_{bi} - z_{b1}) \cdot \theta_x\end{aligned}\quad (1)$$

$$\begin{aligned}\delta_{bzi} &= \delta_z \\ \delta_x &= \delta_{bx1} + (z - z_{b1}) \cdot \theta_y \\ \delta_y &= \delta_{by1} + (z - z_{b1}) \cdot \theta_x \\ \delta_z &= \delta_{bzi}\end{aligned}\quad (2)$$

When we assume the shaft is motioned like rigid body. Following equilibrium equation can be derived. From the three direction equilibrium of the spindle, equation (3) is derived.

$$\begin{aligned}\sum_i^n f_{bx,i} &= \delta_{bx1} \sum_{i=1}^n K_{bx,i} + \theta_y \sum_{i=1}^n K_{bx,i} (z_{bi} - z_{b1}) - \sum_j^m f_{ex,j} \\ \sum_i^n f_{by,i} &= \delta_{by1} \sum_{i=1}^n K_{by,i} + \theta_x \sum_{i=1}^n K_{by,i} (z_{bi} - z_{b1}) - \sum_j^m f_{ey,j} \\ \sum_i^n f_{bz,i} &= \delta_z \sum_{i=1}^n K_{bz,i} - \sum_j^m f_{ez,j}\end{aligned}\quad (3)$$

Where, the $f_{bx,i}$, $f_{by,i}$, and $f_{bz,i}$ are reaction force of each bearings due to geometrical imperfection of the shaft to x , y , and z direction, respectively. And z_b is location of the bearings to be fixed. From the moment equilibrium of the spindle, equation (4) can be derived.

$$\begin{aligned}\sum_i^n f_{by,i} (z_{bi} - z_{b1}) &= \delta_{by1} \sum_{i=1}^n K_{by,i} (z_{bi} - z_{b1}) + \theta_x \sum_{i=1}^n K_{by,i} (z_{bi} - z_{b1})^2 \\ &\quad - \sum_i^n (f_{bx,i} - K_{bx,i} \delta_x) y_{bi} - \sum_j^m f_{ey,j} z_{r,j} - \sum_j^m f_{ez,j} y_{r,j} \\ \sum_i^n f_{bx,i} (z_{bi} - z_{b1}) &= \delta_{bx1} \sum_{i=1}^n K_{bx,i} (z_{bi} - z_{b1}) + \theta_y \sum_{i=1}^n K_{bx,i} (z_{bi} - z_{b1})^2 \\ &\quad - \sum_i^n (f_{bz,i} - K_{bz,i} \delta_z) x_{bi} - \sum_j^m f_{ex,j} z_{r,j} - \sum_j^m f_{ez,j} x_{r,j}\end{aligned}\quad (4)$$

From, the equations (1)~(4), equation (5) can be derived. The first 5X5 matrix of left term means stiffness matrix of the spindle and two summed matrix of right side mean reaction force of the

bearing and external force from torque transmitter, respectively.

$$\begin{bmatrix} -\sum_{i=1}^n K_{ax,i} & 0 & 0 & 0 & -\sum_{i=1}^n K_{ax,i}(z_n - z_{a,i}) \\ 0 & -\sum_{i=1}^n K_{ay,i} & 0 & \sum_{i=1}^n K_{ay,i}(z_n - z_{a,i}) & 0 \\ 0 & 0 & -\sum_{i=1}^n K_{az,i} & 0 & 0 \\ 0 & \sum_{i=1}^n K_{ax,i}(z_n - z_{a,i}) & \sum_{i=1}^n K_{ay,i} \cdot y_{a,i} & -\sum_{i=1}^n K_{ay,i}(z_n - z_{a,i})^2 & 0 \\ -\sum_{i=1}^n K_{ax,i}(z_n - z_{a,i}) & 0 & \sum_{i=1}^n K_{ax,i} \cdot x_{a,i} & 0 & -\sum_{i=1}^n K_{ax,i}(z_n - z_{a,i})^2 \end{bmatrix} \begin{bmatrix} \delta_{ax} \\ \delta_{ay} \\ \delta_z \\ \theta_x \\ \theta_y \end{bmatrix} = \begin{bmatrix} \sum_{j=1}^m f_{ax,j} \\ \sum_{j=1}^m f_{ay,j} \\ \sum_{j=1}^m f_{az,j} \\ -\sum_{j=1}^m f_{bx,j}(z_n - z_{b,j}) + \sum_{j=1}^m f_{bz,j} \cdot y_{b,j} \\ \sum_{j=1}^m f_{bx,j}(z_n - z_{b,j}) - \sum_{j=1}^m f_{bz,j} \cdot x_{b,j} \end{bmatrix} + \begin{bmatrix} \sum_{j=1}^m f_{cx,j} \\ \sum_{j=1}^m f_{cy,j} \\ \sum_{j=1}^m f_{cz,j} \\ -\sum_{j=1}^m f_{dx,j} z_{r,j} + \sum_{j=1}^m f_{dx,j} y_{r,j} \\ \sum_{j=1}^m f_{dx,j} z_{r,j} - \sum_{j=1}^m f_{dx,j} x_{r,j} \end{bmatrix} \quad (5)$$

The reaction force of the bearing can be defined as equation (6).

$$f_e(\theta) = \sum_{k=1}^{\infty} TF(2\pi \cdot k) (a_k \cos 2\pi \cdot k + b_k \sin 2\pi \cdot k) \quad (6)$$

The meaning of equation (6) is Fourier series of the profile on bearings surface and TF is influence coefficient of bearings due to bearing type such as angular contact ball, hydrostatic, or aerostatic bearings [1]. Once we get the displacement of the shaft at bearing 1, we can calculate any error motion of the spindle from the equation (2).

CONFIGURATION OF A HYDROSTATIC SPINDLE

To evaluate the suggested estimation method, a hydrostatic spindle(table) is used for a test bench. 4 Pad of journal and thrust hydrostatic bearings are located in bottom of the spindle. A frameless D.D. motor is used as driver of the spindle and precision hollow through shaft encoder is located on the top of the shaft as shown in Figure 2. The maximum speed of the

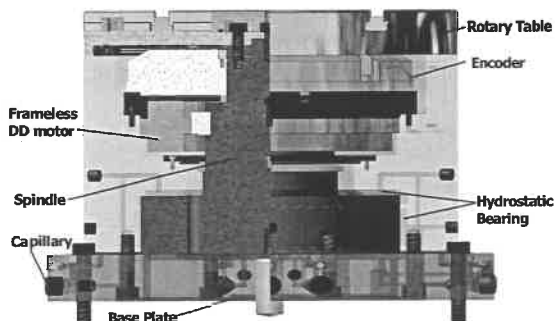


FIGURE 2. Configuration of the hydrostatic bearings rotary table

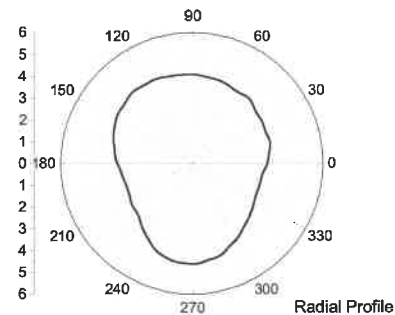


FIGURE 3. Profiles of the shaft for hydrostatic journal bearings

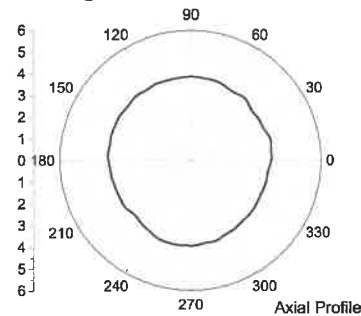


FIGURE 4. Profiles of the shaft for hydrostatic thrust bearings

spindle is 50 rpm.

To calculate the error motion of the spindle, profiles of the shaft for the journal and thrust bearings were measured from the roundness tester. Figure 3 and 4 shows the profiles for the hydrostatic journal and bearings, respectively.

ERROR ANALYSIS

Figure 5 shows the influence coefficient of a 4-pad hydrostatic bearing and it also shows the dominant frequencies. Every amplitude was not same as shown in the figure because of

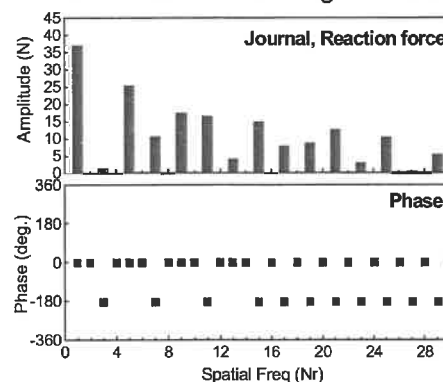


FIGURE 5. Influence coefficient of 4-pad hydrostatic bearing

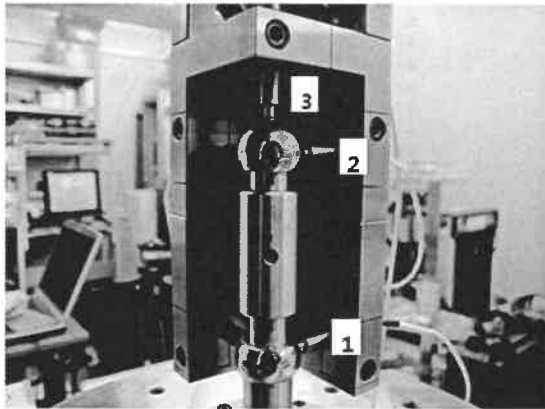


FIGURE 6. Error motion analysis system on the table

averaging effect of the hydrostatic bearings. The spatial frequency indicates k of equation 6. Most of frequency shows cancelling out except dominant frequencies [2].

Effects of bearings

The suggested method was applied to hydrostatic spindle shown in Figure 2. The profile error(roundness) of bearing parts are shown in figure 3 and 4.

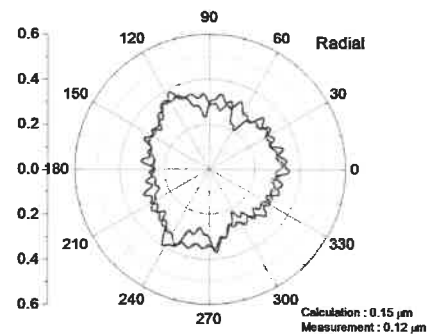
The error motion of the spindle are measured and analyzed by the spindle analysis system with dual master ball. The reversal method [3] is applied to separate the error motion of the spindle from the measured displacement.

Figure 7 shows the results of rotational errors of spindle from equation (5), when the influence coefficient of the figure 5 is applied.

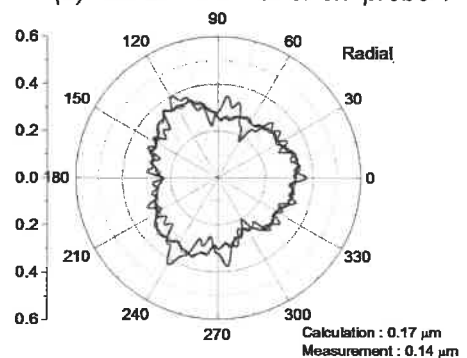
Figure 7 shows the results of a rotational errors of the hydrostatic oil bearing spindle. As shown in the figure, it shows the small asynchronous errors and smoothed motion errors. The advantage of the suggested methods is that the easy extension of the model. Usually, the error model of spindle supported by typical bearing is not easy to change according to change of bearing. But, the suggested influence coefficient method is easy to check the effect of changed bearings. As shown in figure 3, the dominant frequencies of the profiles on journal bearing are 2nd and 3rd order of the profiles, but from the influence coefficient of the bearings 2nd order was cancelled out and 3rd order are remained and 3 robe like spindle errors can be measured as shown in Fig. 7 (a) and (b).

Effects of Driving Components

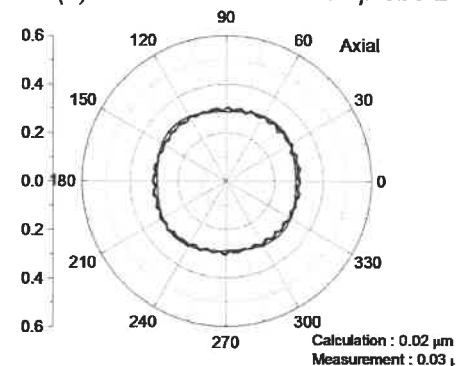
The effect of driving component also easy to use, when the suggested method is applied. Direct



(a) Radial motion error on probe 1



(b) Radial motion error on probe 2



(c) Axial motion error on probe 3

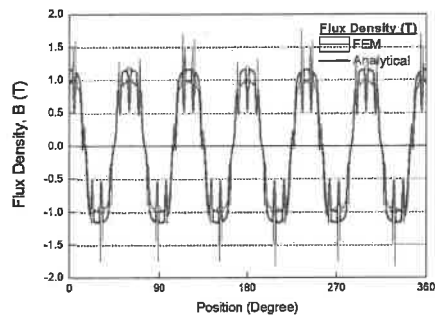
FIGURE 7. Comparison of the motions between estimated and measured errors

drive (D.D.) motor was used in the hydrostatic spindle and the force model of the D.D motor can be derived from magnetic flux model.

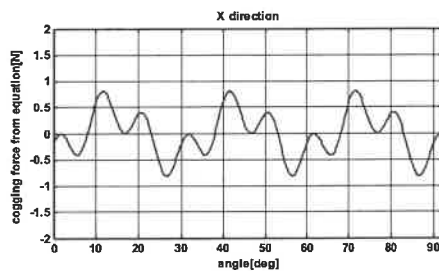
$$Fe_{12x} = M_{12x} \sin(12\theta + \phi_{12x}) \quad (7)$$

$$Fe_{36x} = M_{36x} \sin(36\theta + \phi_{36x}) \quad (8)$$

The analytical model was compared with FEM analysis and experimental results and figure 8 shows the analytical results of the magnetic flux and radial deviations of the motors. This model is applied to Equation (5) and small ripples of



(a) Magnetic flux density of the D.D motor



(b) Radial force estimation

FIGURE 8. Radial force of the D.D. motor

the error motions of the figure 7 is owing to the effects of the force of figure 8

CONCLUSIONS AND FURTHER STUDY

In this paper, an estimation method for the five degree of freedom (5 D.O.F) error motions of the spindle is suggested. To estimate the error motions of the spindle, waviness of shaft and bearings, external force model was used as input data. And, the estimation models are considering geometric relationship and force equilibrium of the five degree of the freedom. From the simulation results, we confirmed that this method has advantage of exact and fast calculation for the rotational errors of spindle.

The effect of vibration in high speed motion regarding vibration response of driving force and unbalance model will be added.

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DEVELOPMENT OF A LASER-GUIDED DEEP-HOLE MEASUREMENT SYSTEM WITH HIGH-PERFORMANCE MECHANICAL ACTUATORS

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INTRODUCTION

It is essential to accurately measure the diameter, roundness, cylindricity, and straightness of a deep hole for applications such as the rotation shafts of a jet engine, generator and high-speed train; the cylinder used in plastic injection molding; and the cylinder liner of a ship engine. Existing system has drawbacks in precise measurement of the hole with large length to diameter ratio, in which different measurement devices are required separately.

To evaluate the accuracies of such deep holes using a single device, a laser-guided deep-hole evaluation probe is developed. Expensive piezoelectric actuators are used to control attitude of the probe [1]. It is still difficult to measure the accuracy of holes with depths in the order of a few meters and with diameters varying in the order of a few millimeters, such as the rotating shaft holes of jet engines and high-speed-trains.

The objective of the current study is to design laser-guided probe to satisfy the following two conditions:

- (1) To reduce the production cost and achieve a high measurement accuracy.
- (2) To measure the accuracy of the hole with a depth in the order of a few meters and a diameter varying from 70 to 168 mm.

In this study, a measurement unit is fabricated and its performance is assessed using a lathe. Further, a set of actuators are fabricated and their performance is assessed.

NEW HIGH-PERFORMANCE LASER-GUIDED MEASUREMENT SYSTEM

Figures 1, 2, and 3 show a new laser-guided measurement probe, a measurement system, and a measurement unit that can be adjusted to a hole diameter ranging from 70 to 168 mm. The measurement system mainly controls the attitude of the probe and senses the hole wall.

Probe Attitude Control

The probe attitude (position and inclination) is detected by a laser diode 5 and two CCD cameras 14 and 15 (Fig.2). The attitude is controlled using six actuators such as 3, 4 (Figs.1 and 2) [2]. The six actuators are positioned circumferentially and longitudinally (Fig.1). The longitudinal space is necessary for placing the actuators that consist of worms, worm wheels, and stepping motors.

Hole Wall Sensing

The hole wall is scanned spirally by a stylus which is a component of the measurement unit. The movement of the stylus is detected by a laser interferometer 12 (Fig.2).

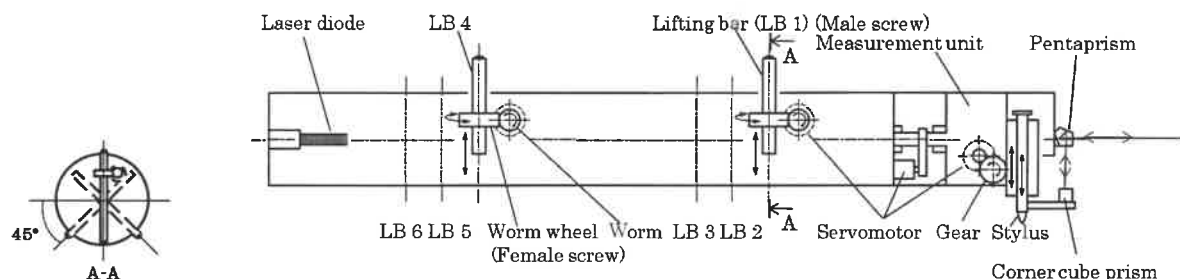


FIGURE 1. New laser-guided measurement probe adjustable to hole diameters ranging from 70 to 168 mm.

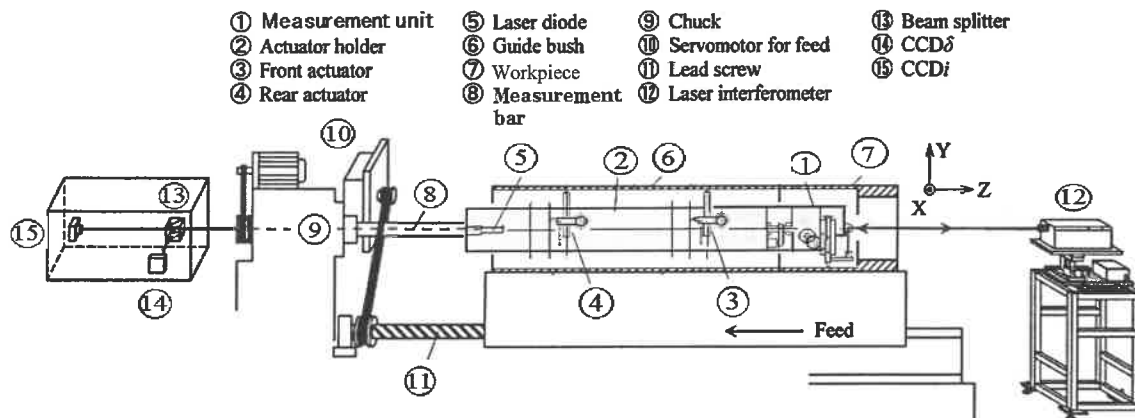


FIGURE 2. New laser-guided measurement system.

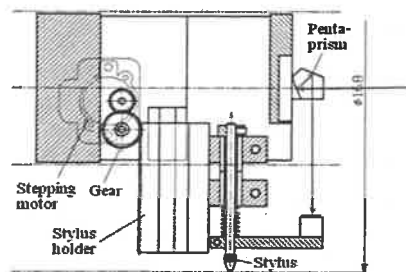


FIGURE 3. Measurement unit.

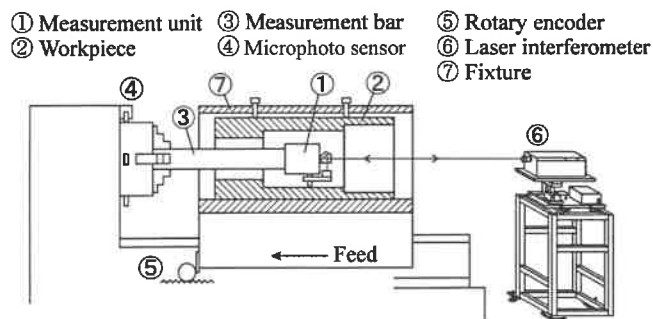


FIGURE 4. Measurement system.

MEASUREMENT PRINCIPLE

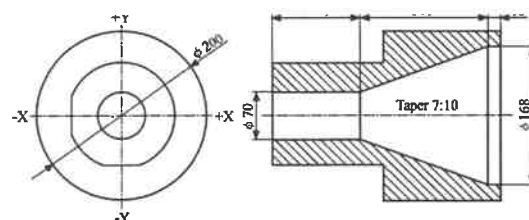
The measurement unit consists of a pentaprism and a corner cube prism fixed to the stylus (Fig.3). When the hole diameter varies considerably, hole accuracy is measured by pushing the stylus holder toward the hole wall by a stepping motor and gear. A laser interferometer 12 is placed in front of the probe (Fig.2). The laser beam is incident on the pentaprism and deflected to a corner cube prism. The reflected beam from the corner cube prism returns via pentaprism to the interferometer. He-Ne laser is used: generating power 1mW, wavelength 632.8nm. The resolution of the interferometer is 0.01 μm .

PERFORMANCE OF MEASUREMENT UNIT

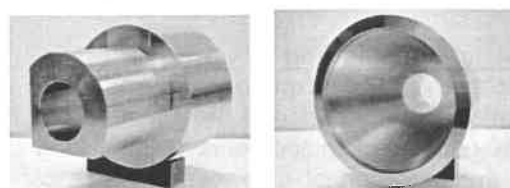
Performance of the measurement unit is examined by measuring accuracy of a hole fixing at the free end of a measurement bar (Fig.4).

Experimental Method

The hole accuracy is measured by using a lathe (Fig.4) [3]. A workpiece (duralumin: JIS-type A2017-T4) set on a carriage is fed to the probe and the measurement unit is rotated. By the combination of these two motions, the stylus



(a) Dimension of workpiece (duralumin)



Left side view Right side view

(b) Photograph of workpiece

FIGURE 5. Shape of workpiece.

spirally scans the wall of the hole. The rotational speed is 15 min^{-1} and the feed rate is 0.94 mm/rev.

Figure 5 shows the shape of workpiece is shown. The hole diameter varies from 70 to 168 mm, as shown in Fig.5(a). The hole shape is bell-

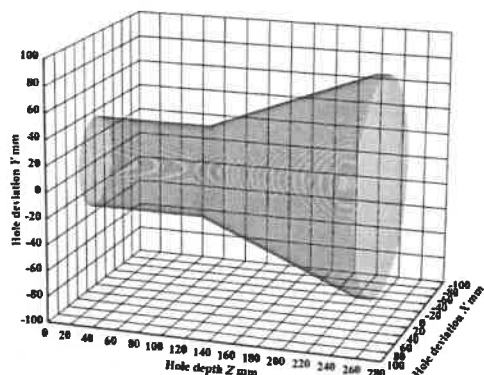


FIGURE 6. Spiral measurement of hole shape.

mouthed, as shown in Fig.5(b). It is very difficult to accurately measure the shape using a conventional measurement device.

Experimental Results

Figure 6 shows the hole shape measured spirally by the measurement unit. The hole shape corresponds well with the design and photograph of the hole shown in Figs.5 (a) and (b). The roundness measured by both the measurement probe and the roundness tester are shown at depths of 55 and 245 mm in Figs.7 (a) and (b), respectively.

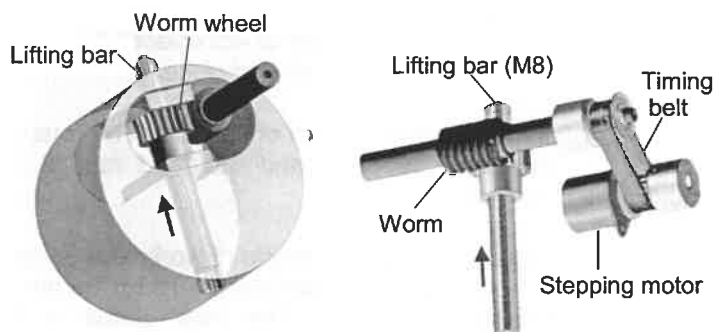
The roundness measured by the measurement unit corresponds well in shape and value to that measured by a roundness tester.

PERFORMANCE OF ACTUATOR

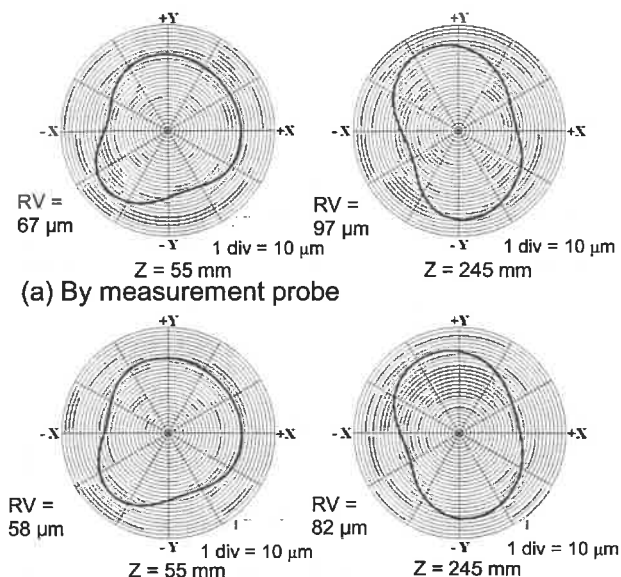
Figure 8 shows an actuator that is fabricated to test its feasibility in the measurement probe. The lifting bar is driven by the stepping motor (CSA-UA56D1, Shinano Kenshi) through the worm and worm wheel, with a stroke of 50 mm.

Experimental Method

Figure 9 shows experimental apparatus for examining performance of the actuator. Loading of weight, the displacements of the lifting bar, and the detection of displacements are carried



(a) Set in actuator holder (b) With stepping motor
FIGURE 8. Actuator with a worm and worm wheel system.



(b) By roundness tester

FIGURE 7. Roundness measured by the measurement probe and roundness tester.

out using a steel measurement bar with a diameter of 12 mm. The actuator is fixed with a vice. The lifting bar is moved upward and downward by the stepping motor. A force F N acts on the lifting bar when a dead weight is loaded on to the measurement bar. The displacement of the lifting bar is measured using the above described laser interferometer via the measurement bar.

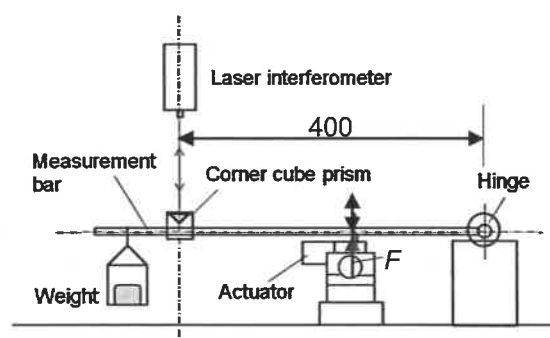
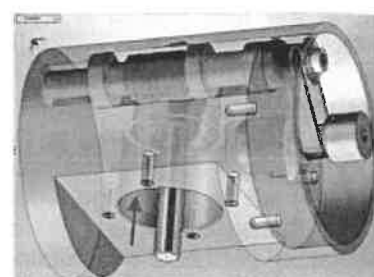


FIGURE 9. Experimental apparatus for examining performance of the actuator.



(c) Working condition

Experimental Results

Figure 10 shows the variations in the displacement of the lifting bar with respect time, with force F acting on the lifting bar. The solid and dashed lines indicate upward and downward displacements of the lifting bar, respectively.

The displacement increases linearly with respect to time despite the strength of the lifting force F . The achievement of the same displacement is delayed with the increase in force F . This is because the rotational speed of the stepping motor decreases with an increase in the force F . S light deviation from the linear line of displacement because of fabrication error in the actuator. In particular, the large clearance between the worm and the worm wheel causes a nonlinear increment of the displacement.

The maximum lifting force F of the actuator is 163 N. During the experiments, the rotational speed of the stepping motor was 750 rpm. However, This speed can be increased up to 3000 rpm.

COMPARISON WITH PIEZOACTUATOR

Thus far, piezoelectric actuators have been used for controlling the probe attitude. Figure 11 shows the relationship between the displacement and the voltage impressed on the actuators [2]. The maximum lifting force F and the displacement are 160 N and 0.35 mm, respectively. In the mechanical actuator used in our experiments, there is no change in the displacement due to force F and a hysteresis error does not occur. The cost of the mechanical actuator can be reduced considerably. A wide displacement stroke in the mechanical actuator can be achieved successfully.

CONCLUSIONS

A laser-guided measurement system with a probe that controls its attitude by mechanical actuators is designed. Further, a measurement unit and an actuator are fabricated and their performances are assessed. It is experimentally found that the measurement unit and the actuator can be used practically. The result shows that this system can be implemented in various measurement applications.

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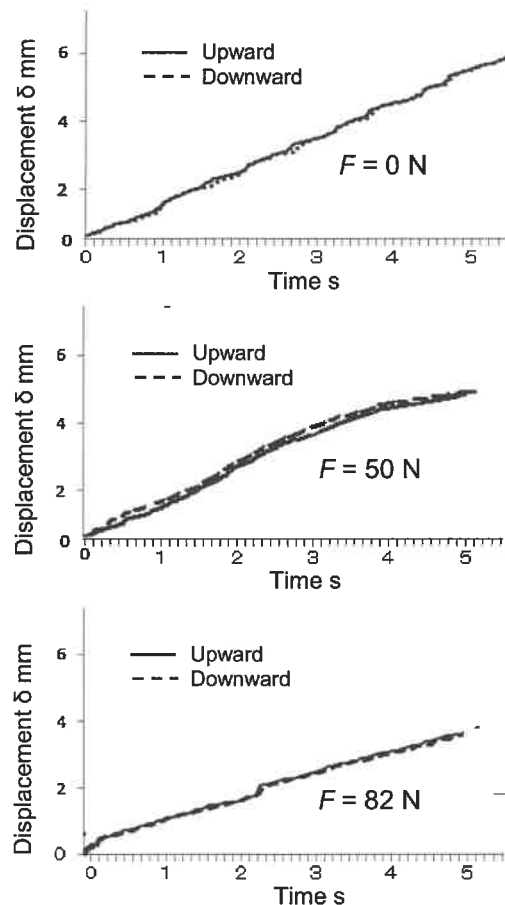


FIGURE 10. Variation of displacement with time.

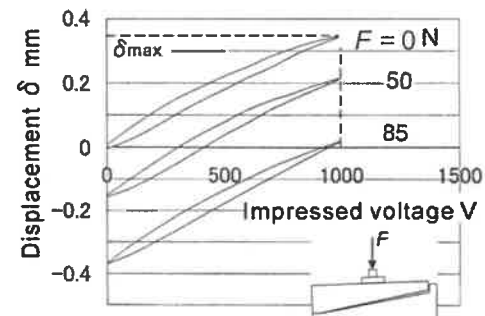


FIGURE 11. Relationship between displacement and voltage impressed on the actuator.

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MULTI-PROBE ERROR SEPARATION TO ELIMINATE THE INFLUENCE OF BALL BEARING SPINDLE MOTION IN PROCESS MEASUREMENTS

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INTRODUCTION

Process measurements from a machine utilizing roller bearing spindles can be dominated by the asynchronous motion introduced by the rolling bearing elements[1]. This paper demonstrates a technique for using multi-probe error separation for removing this motion and the synchronous shape of the spindle in order to monitor the process of interest.

BACKGROUND

Ball bearing spindles are used in many machines of importance in precision engineering, for example, milling machines and surface or cylindrical grinding machines. It is desirable to measure the process variables of these machines for purposes such as determining the condition of tools or grinding wheels [1] [2]. Common approaches to measuring process parameters in this sort of machine include using acoustic emission [3] or accelerometers. With this sort of measurement, the entire process is captured, and any measurements will be dominated by bearing motion and spindle eccentricity. This results in the difficulty of finding small changes in a noisy signal. However, the signal amplitude at frequencies synchronous to the spindle speed is significantly lower.

The synchronous error motion of a ball bearing spindle being used in many operations has components resulting from external forces. For example, a milling machine spindle will have synchronous error motion as a result of cutting forces. These forces will occur at integer multiples of the spindle speed, for example, at the number of cutting faces involved in cutting at one time. These external forces will result in error motion of the spindle. Multi-probe spindle error motion separation is a natural way of extracting this error motion[2].

APPROACH

In order to separate spindle error motion from the eccentricity of the spindle, the multiprobe method is utilized. A schematic of the multiprobe technique is shown in 1.

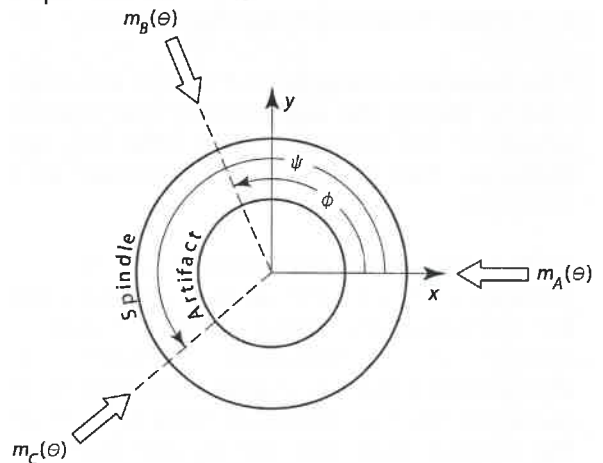


FIGURE 1. Multiprobe error separation measurements

This requires simultaneously sampling three sensors measuring the spindle circumference. The angle between the sensors is optimized to resolve the spindle shape to as high a frequency as desired within the angular resolution of the sensors [4]. The sensor mounting nest is shown in Figure 2 .

A software program was written in the Matlab programming language to perform the required data acquisition and processing to perform multiprobe separation. To demonstrate the proper functioning of this software, it is expedient to perform a marker test. This test consists of marking a line on the spindle with a permanent marker. The ink from the permanent marker affects the reading from the sensors in such a way that it appears to be a large outward excursion in the spindle circumference. If properly separated, the effects of

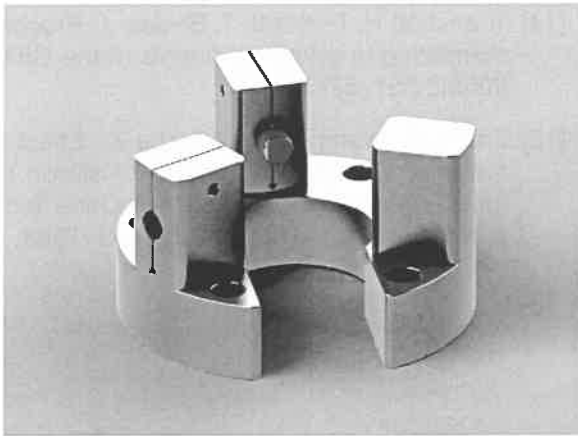


FIGURE 2. Capacitance probe holding fixture

the marker will be limited to the spindle shape, and not the spindle error motion. Polar plots of the raw data from such a test are shown in Figure 3.

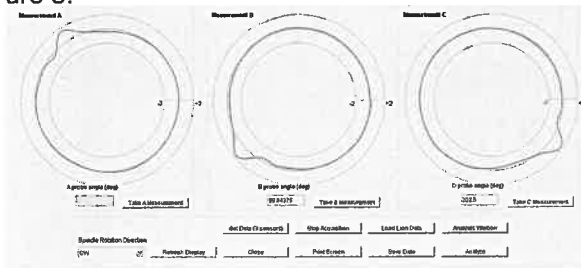


FIGURE 3. Raw data from 3 sensors for multiprobe error separation scheme

The results of the separation are shown in Figure 4. Inspection of this figure shows that the marker is only found on the spindle (workpiece) plot.

The advantage to the multiprobe technique is that the error motion separation can be performed on-line. While the technique requires data in increments of a complete evolution of the spindle, there are a large number of manufacturing processes where this is an acceptable rate of analysis.

EXPERIMENT

A ball bearing spindle with a cutting tool was instrumented for multiprobe analysis. Three capacitance probes were arranged at suitable angles around the spindle. The angular position of the spindle was measured using a 512 count optical encoder. The signal from the capacitance probes were acquired using a general-purpose data acquisition board hosted in a desktop PC. The ca-

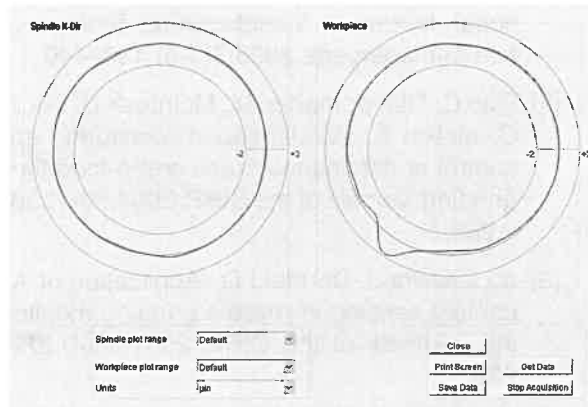


FIGURE 4. Results from multiprobe separation scheme showing error motion and spindle shape

pacitance probe signals were sampled at each negative-going transition of the optical encoder.

The spindle and cutting tool used in this experiment is such that the cutting forces can be adjusted via the positioning of the tool. The experiment consisted of moving the tool to a given position, performing multiprobe separation, and solving for the synchronous error motion of the spindle. The amplitude of the error motion at the frequency corresponding to the number of cutting faces on the tool was recorded as a function of tool position. This relationship is shown in Figure 5.

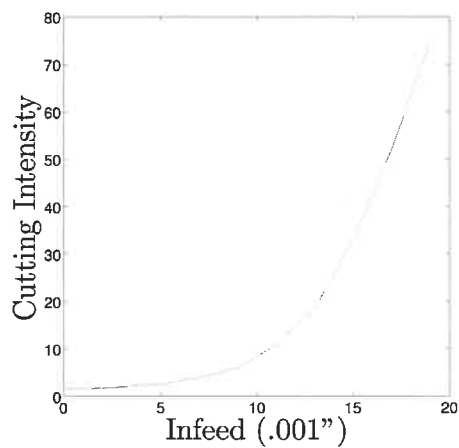


FIGURE 5. Cutting intensity at the frequency corresponding to the number of cutting faces

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DESIGN, MANUFACTURE, AND TESTING OF NANOMETER-LEVEL SPINDLE METROLOGY TESTBALLS

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ABSTRACT

Nanometer-level spindle metrology requires a suitable artifact, an accurate sensor, stiff fixturing, and a low-noise data acquisition and display system [1]. A commonly used artifact for this purpose is a lapped sphere, referred to here as a testball. This paper discusses spindle metrology and the aspects of testball design, ball lapping, metrology, and calibration using multiprobe error separation.

SPINDLE METROLOGY BACKGROUND

Schlesinger is credited with developing the first systematic approach to the testing of machine tools—including spindle testing. In his book *Testing Machine Tools*, he was the first to describe a method to measure spindle accuracy using contact displacement indicators and a cylindrical test mandrel or testball [2]. Schlesinger pointed out that artifact out-of-roundness and spindle error motions are measured simultaneously so the artifact form error must be minimized.

Thrust generated polar plots at operating speeds with an oscilloscope and non-contact sensors [3]. The Thrust method produces rotating sensitive radial error motions applicable to milling and boring [4]. Bryan, Clouser and Holland extended this approach to measure lathes with fixed sensitive direction error motions [5]. Bryan et al. suggest a check for the problem of artifact out-of-roundness by orienting the sphere in several angular positions relative to the spindle rotor and observing changes in the measurement. They point out, however, that as the spindle and artifact accuracies approach one another, efforts must be made to distinguish and separate each error.

The reversal techniques described by Donaldson and Estler theoretically provide the simplest method for separating spindle error motion from artifact form error [6,7]. In a production environment, however, measurement accuracy of these techniques suffers from the difficulty of exact reversal of the ball [8]. Particular care must be taken to accurately remount the testball 180° without influencing form error, which can be difficult when testing at high speed. Any change in the testball between the tests, from a fingerprint or dust, is split evenly between the spindle and artifact.

Whitehouse developed a multiprobe technique which avoids some of the complications of reversal [9]. This method requires accurate positioning and alignment of the sensors. Asymmetric angular spacing of the sensors is selected to minimize harmonic suppression and provide results comparable to Donaldson reversal [10].



FIGURE 1. General-purpose single and dual ball artifacts.



FIGURE 2. Monolithic testball for testing at-speed.

TESTBALL DESIGN

Some general purpose single and double testballs are shown in Figure 1. This design is well-suited for many types of spindles, but needs refinement for measurements repeatable to the nanometer-level.

To resolve nanometers at high speeds, provision for balancing, stiff and repeatable mounting interface, and uniform circular stiffness are of critical importance. The flange-mounted ball in Figure 2 is designed for measuring spindle error motion on the order of 10 nanometers. The testball is manufactured from 440C stainless steel bar stock, hardened to 60 Rc and stabilized. The testball is securely held to the spindle by six mounting bolts. The mounting flange is roto-ground flat to better than 0.25 μm . Six tapped holes are added to the flange to enable balancing.

TESTBALL MANUFACTURE

After turning, the testball is ground in a sphere generator with an air bearing spindle. A 4B BLOCK-HEAD[®] air bearing spindle is mounted horizontally and the testball is aligned to the axis of rotation. The grinding spindle sweeps a 25 mm path over the pole of the rotating ball and past the equator. The air bearing work spindle provides accurate rotation which result in sub-micron form errors around the critical equator of the ball.

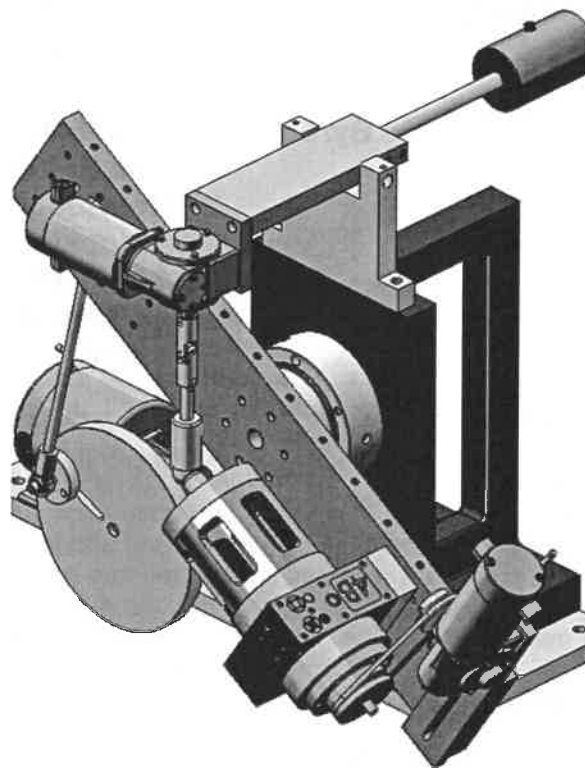


FIGURE 3. Purpose-built testball lapping machine.

A purpose-built lapping machine shown in Figure 3 finishes the job. This machine uses two spindles and a cup lap. A ball is mounted to a model 4B BLOCK-HEAD[®] air bearing spindle that rotates continuously. The mounting fixture is designed so that the center of the ball intersects the axis of rotation of a model 4RT tapered roller bearing spindle. The 4RT acts as a pivot for a linkage that oscillates the inclination angle of the ball relative to the lap. A non-influencing coupling is used to seat the cup lap on the ball. The design provides variable cup lap speed and variable ball rotation speed. The cup lap contact force is set using a counterweight. Fifteen micrometer diamond compound is used for initial rough lapping; one micrometer lapping compound is used for final finishing.

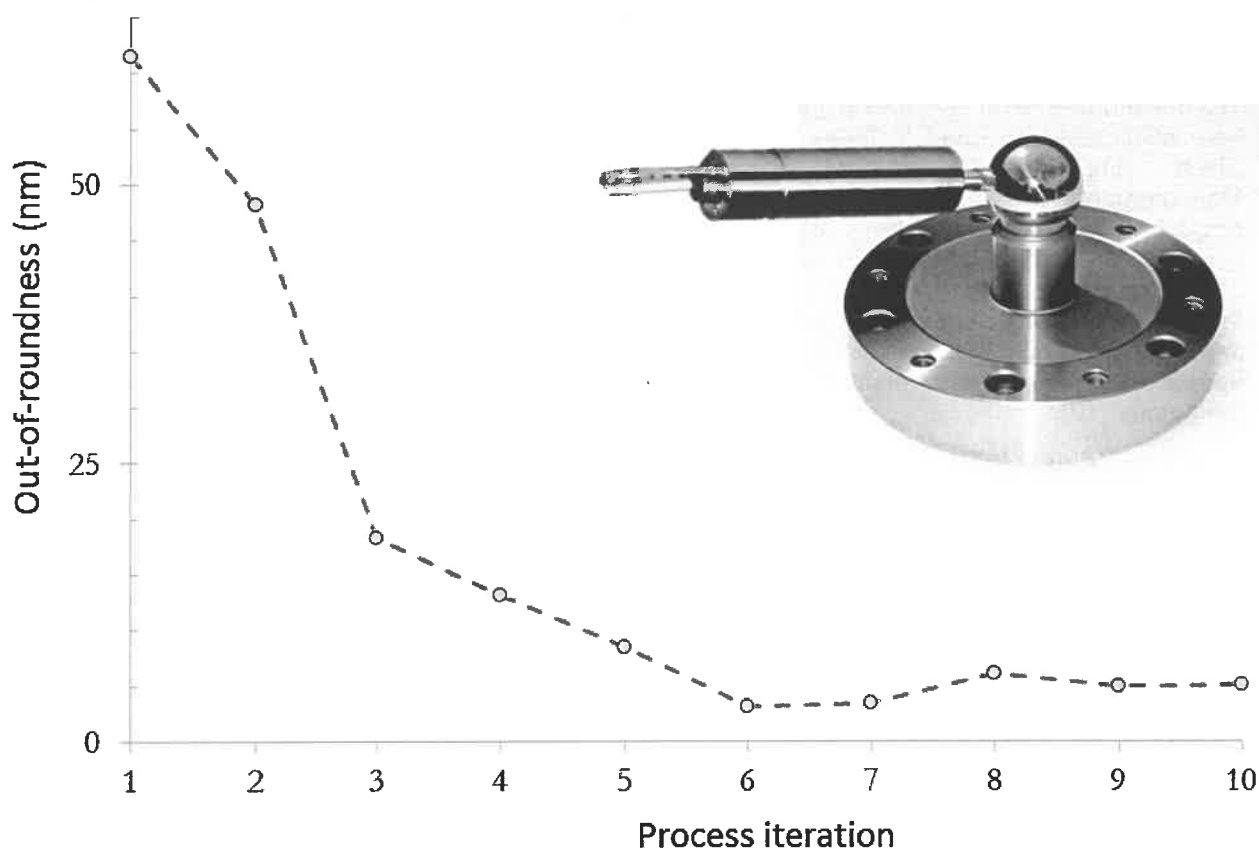
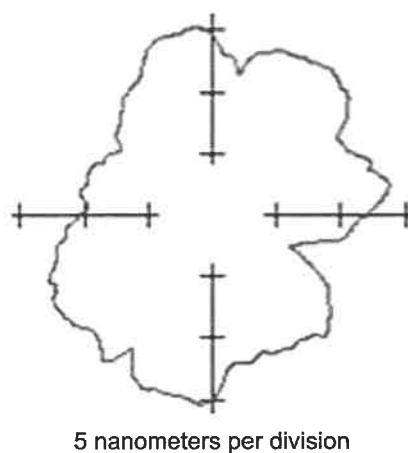


FIGURE 4. Improvement in ball roundness measured with an air bearing contact sensor.

TESTBALL METROLOGY

Despite the ability to perform error mapping, it is still necessary to minimize testball form error for the sake of improving the signal-to-noise ratio. To achieve form errors on the nanometer-level, out-of-roundness measurements are used as feedback during lapping. A history of the out-of-roundness for a typical ball is shown in Figure 4. The purpose-built roundness checker uses a 4R BLOCK-HEAD® air bearing spindle to rotate the part and an air bearing contact sensor with a diamond tip to measure out-of-roundness (Lion Precision C-LVDT). Adjustments to the ball rotation speed, lap speed, lap time, abrasive size, and lap force can be made with intermittent metrology feedback.

After lapping is complete, calibration of the testball is done using multiprobe error separation. Accurate calibration assigns the contributions of spindle and artifact in the correct proportion.



5 nanometers per division

FIGURE 5. Out-of-roundness of the 25 mm diameter testball is 9 nm at the equator as measured with a capacitive sensor at 10,000 RPM.

Several factors play a role in proper error separation including sensor positioning, data acquisition triggering with a minimum of angular

uncertainty, stiff fixturing and thermal stability [1].

The multi-probe error separation method used here eliminates the need to index the artifact which simplifies the testing procedure. Measurements are made sequentially with a single probe to reduce sensitivity differences. In some cases, capacitive sensors are used for calibration to capture the effect of spot size on the out-of-roundness measurement. This implementation of the multiprobe technique agrees with Donaldson reversal to less than a nanometer [12].

CONCLUSIONS

By using modern capacitive sensors, high quality data acquisition systems, precision engineered structures, and appropriate signal processing techniques, it is now feasible to manufacture and measure spindles and artifacts with sub-nanometer repeatability and nanometer reproducibility.

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ANISOELASTIC SOURCES OF ERROR MOTIONS

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ABSTRACT

For rotors, the consequences of non-uniform circular elasticity are twice-per-revolution error motion for fixed sensitive direction and major and minor centering errors for rotating sensitive direction. The case is reversed for stators and stator support structures. These effects apply to bearings, spindles, chucks, workpieces, machine tools, metrology equipment, gyroscopes, and flyrods. Examples are given for horizontal and vertical axes, with illustrations of specific imperfections. Distinction is made between error motion and compliance so that unambiguous specifications can be constructed and error sources identified.

BACKGROUND

Structures designed for symmetry will nevertheless at some level demonstrate non-uniform compliance under load. Hand-built flyrods are rotated horizontally to find the plane of maximum stiffness so that the line guides can be installed in the plane of the "spine", gyroscope designers minimize drift by designing for isoelastic bearings and support structures, spindle designers aim for symmetrical bearings, rotors, and stators, builders of machine tools and metrology equipment aim to minimize the effects of load variations and manufacturers of precision parts are interested in searching for root causes of form errors. All these relate to behavior of structures under load and the relationship between compliance and error motion.

EULER'S PRINCIPAL AXIS THEOREM

Any shape from a round rod to random shapes will have a stiffest orientation angle, with the least-stiff orientation at 90°. A well-made rotor shaft may be nominally symmetrical but since nothing is perfect there will always be some effect. The design goal is circular symmetry, including joints. Oval sections are the worst, all other polygons are OK in principle but 2-lobe will always be present to some degree.

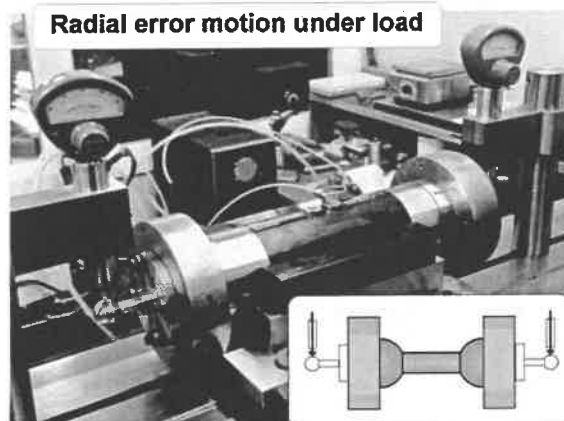


Figure 1, Testing the circular elastic symmetry of a spindle rotor by adding radial loads, circa 1960. Twice-per-revolution radial error motion can originate in the bearings, the connecting shaft, or the bolted joints, (including the test ball attachments).

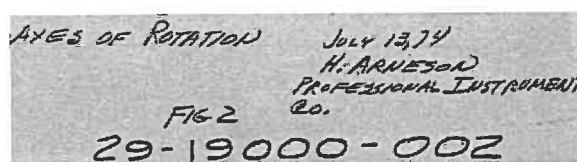


Figure 2. The effect of non-rotating loads on structures having imperfect circular elastic symmetry was a subject of correspondence with the original committee charged with developing concepts and terminology for B89.3.4 Axes of Rotation: Methods for Specifying and Testing. [1,2,3,4]

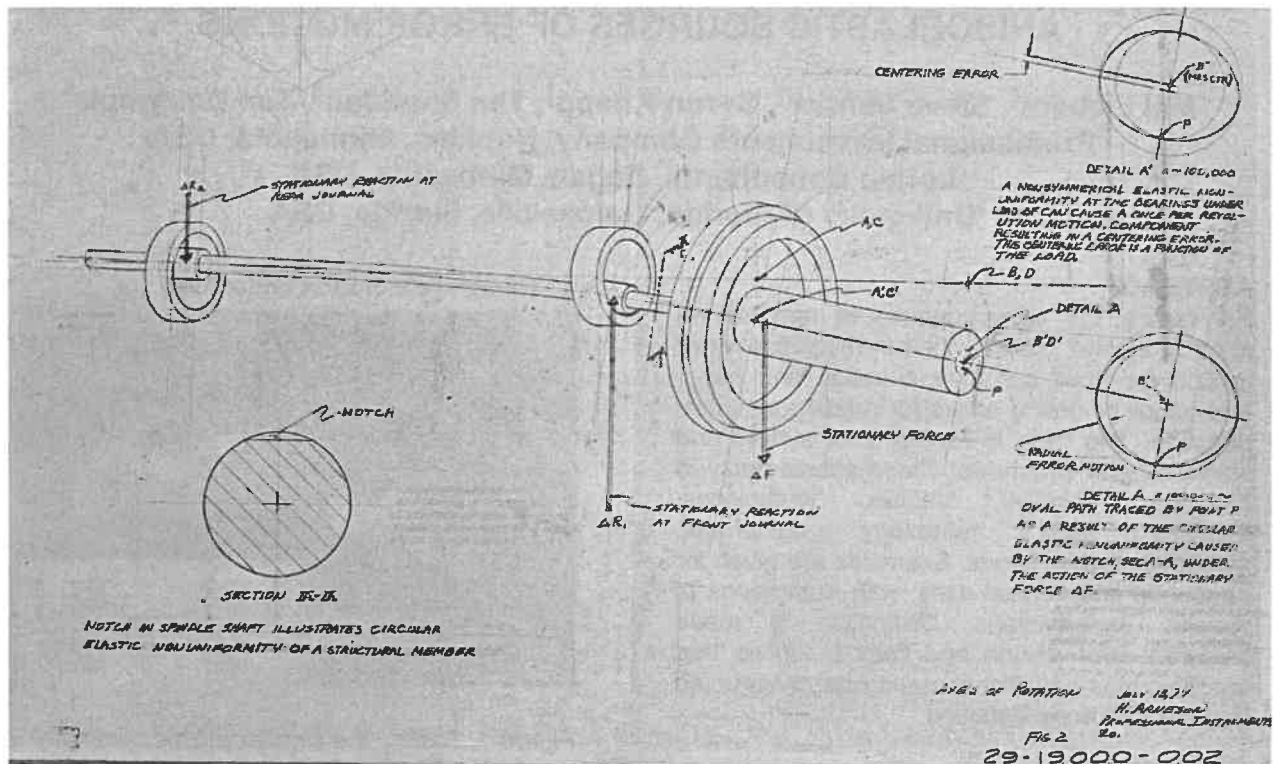


Figure 3. An otherwise perfect spindle will, when under load, have twice-per-revolution radial error motion due to imperfect circular elasticity of its rotor shaft. From the outside looking in at the rotor, the defect is seen as a oval path but from the rotor looking out the effect is seen as a centering error varying in magnitude according to the angular orientation of the point of observation with respect to the notch.

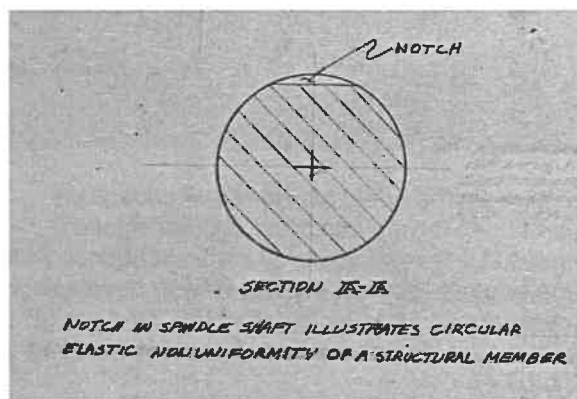


Figure 4. Here the discontinuity is a notch, but it's usual to see keyways, cross-holes, and other design features unintentionally spoiling the symmetry [3,4,5]. At the highest levels of precision, even variations in material can be significant.

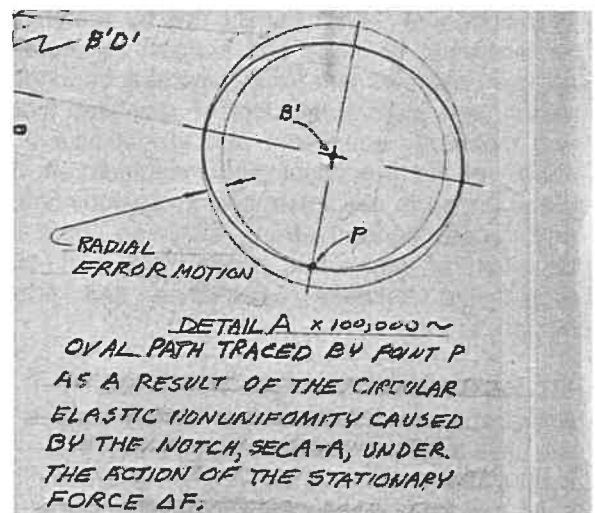


Figure 5. The notch alone is not a problem until a radial load is applied, at which time a twice-per-revolution radial error motion is induced. This is the argument for testing spindles under load and why good designers worship at the shrine of symmetry.

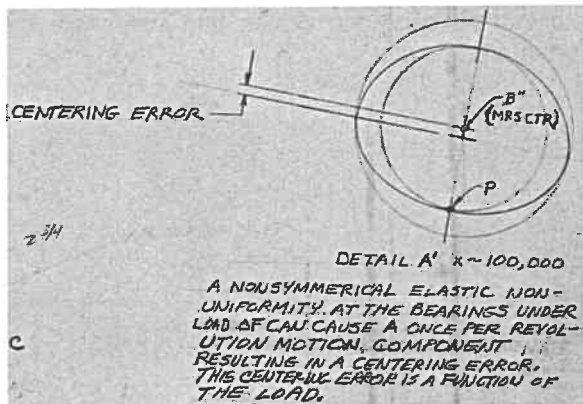


Figure 6. The worst-case centering error is in the soft direction as shown here, but of course a lesser error occurs in the stiff direction when the load changes. Changing the load on the rotor shifts the axis of rotation an amount determined by compliance, (which is not to be confused with error motion).

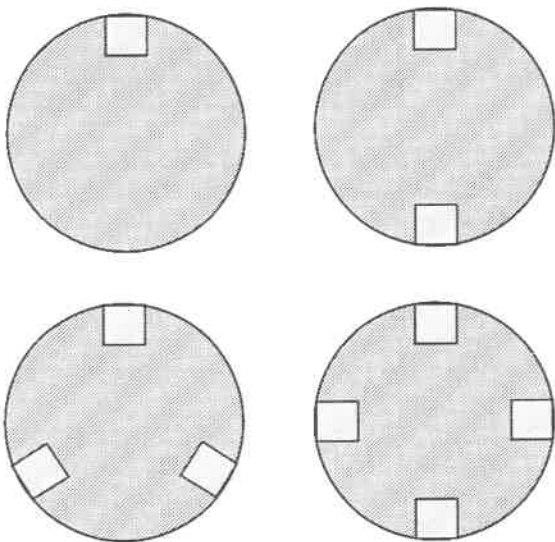


Figure 7. A single discontinuity spoils symmetry and two is worse, but any other number is OK as long as the features are equally distributed. In other words, the only two cases are equal or unequal bending in X and Y; it is not possible to create three-lobe bending by a triangular shaft and Euler's Principal Axis Theorem teaches us not to try.

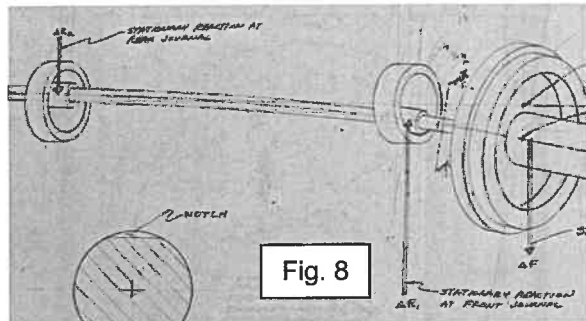


Figure 8. Rotor imperfections are revealed by the steady force of gravity. Stator imperfections are revealed by the rotating force from unbalance. Overhung loads are supported by the front bearing, tilt loads are resisted by the rear bearing and this bends the rotor shaft.

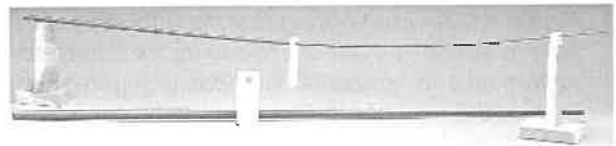


Figure 9. A commercially available "spine finder". Fly rod builders have long known that their rods deflect different amounts at different orientation angles; they don't all agree on placement of the line guides but they usually care enough to test for the effect.

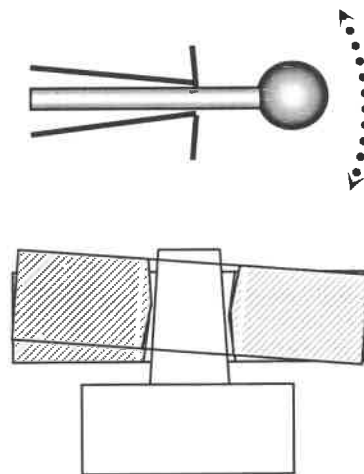


Figure 10. Collets, tapers, and chuck jaws are handy but they rarely grip with perfect uniformity. The result is structural error motion of rotor-mounted hardware, not to be confused with spindle error. Note that the axis of rotation is uninvolved.

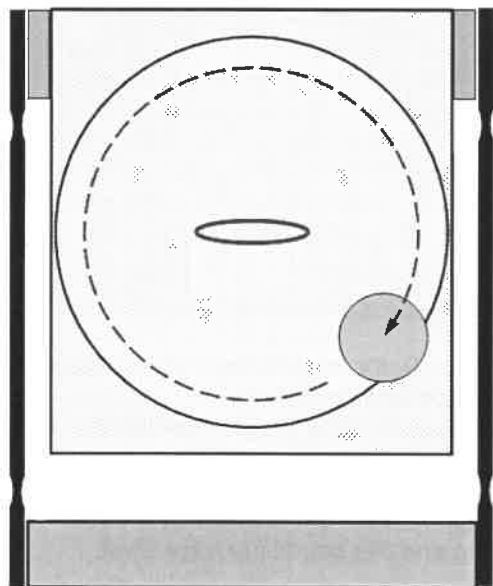


Figure 11. An oval-boring test rig with reed-flexure supported stator. The extreme difference in vertical and horizontal stiffness elongates the path of the axis of rotation when the rotor is unbalanced. The stiffness difference also causes the once-per-revolution displacement to change its orientation angle with respect to the stator as described in Fig. A-20 Effect of Unbalance on Hole Boring [2].

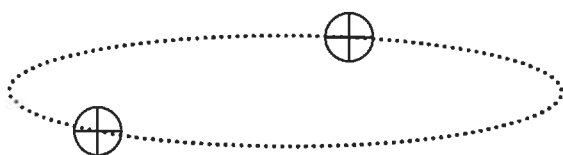


Figure 12. The axis of rotation sways to and fro with the stator but the stator-to-rotor spindle error motion is only affected by anisoelasticity within the spindle stator. The displacement of the axis of rotation is a function of the stiffness of the reeds and of the unbalance.

CONCLUSIONS

Centering errors and twice-per-revolution axis error motions are interrelated but are seen individually depending on whether the point of observation is stationary or rotating. Lack of circular elastic symmetry is a frequent source of this effect. Symmetry is required of the spindle, the spindle support hardware, and the rotor-mounted structures. Anisoelastic structures are

subject to centering errors that vary with the orientation angle.

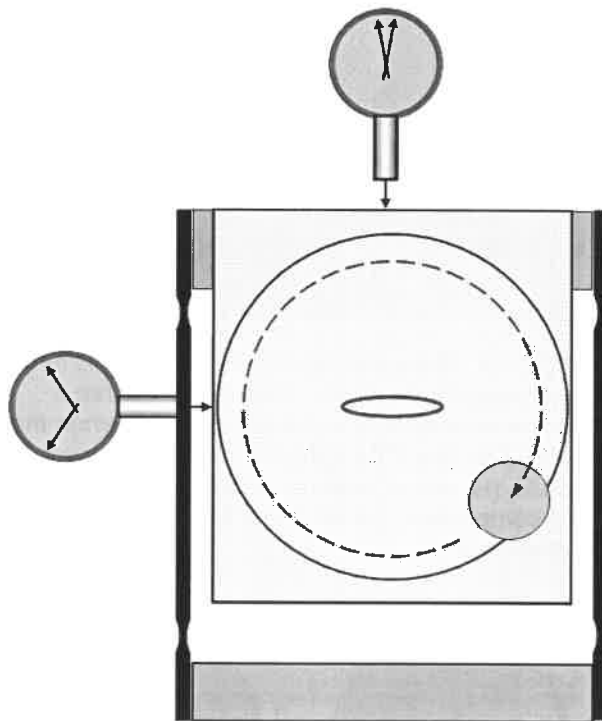


Figure 13. Unbalance creates an oval orbit of the axis of rotation as seen from the base to the spindle stator.

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VERTICAL-AXIS EFFECTS ON SPINDLES

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ABSTRACT

For vertical-axis applications, an added load shifts the average axis line lower and unbalance causes a centering error (or can correct it). Compliance of the spindle bearings lowers the rotor but the axis of rotation maintains its orbit about the axis average line. A radial force tilts the workpiece but not the axis of rotation.

BACKGROUND

There is a long-felt need for an entry-level treatment of the results of load variation for vertical axis applications. Two examples are discussed: A theoretical discussion of the effects of adding axial and radial loads, and distinguishing between compliance and error motion [1]. A workshop example of fine-centering of a test ball by adding weights to shift the rotor.

THEORY

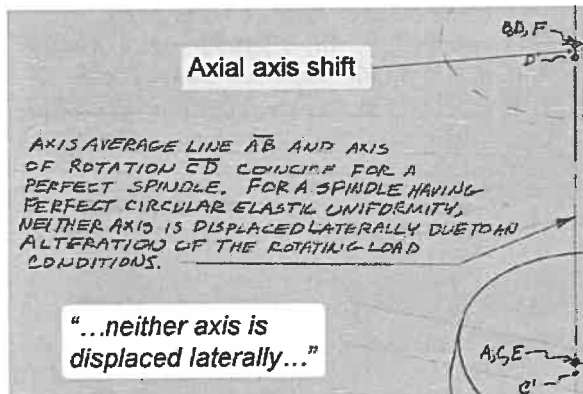


Figure 1. When an off-axis weight is added, points on the axis average line move down by an amount determined by the axial compliance of the spindle bearings. The rotor tilts but not the axis of rotation. A previously centered artifact now must be re-centered.

PRACTICE

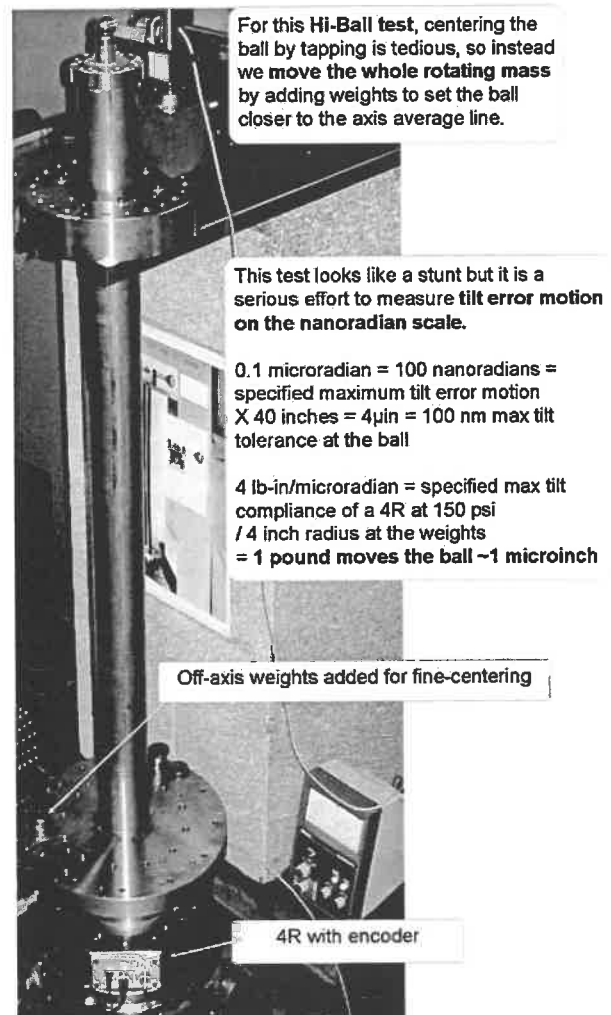


Figure 2. Hi-ball test for tilt error motion, using tilt compliance to adjust TIR.

EFFECTS OF THE ALTERATION OF SYNCHRONOUSLY ROTATING LOADS

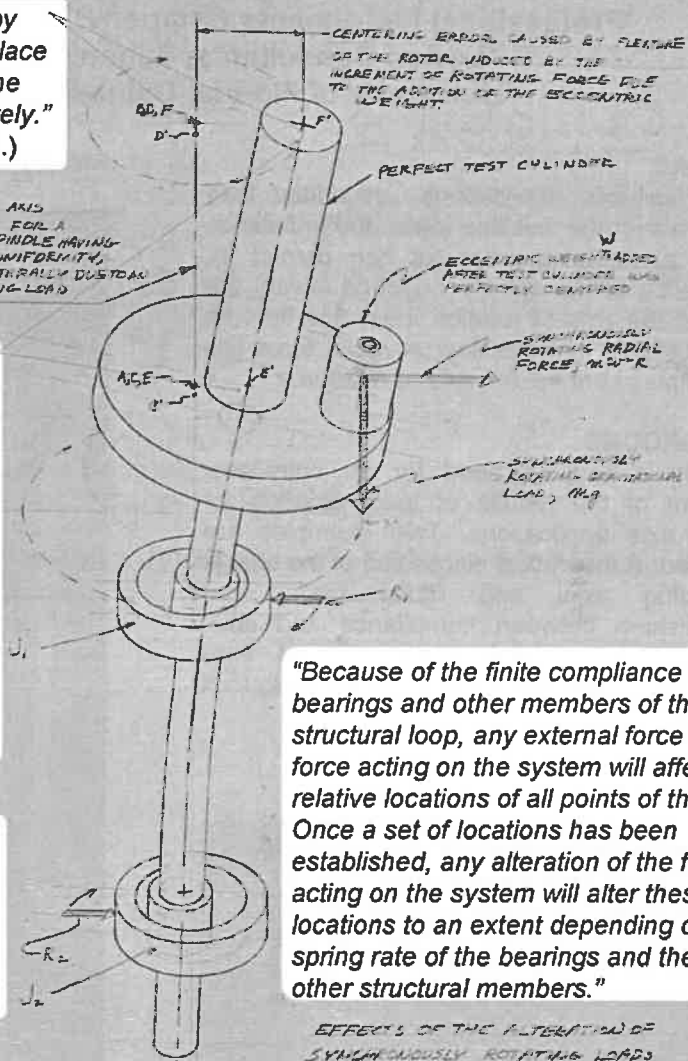
"The elastic deformation caused by the addition of the weight will displace the points E and F of the axis of the test cylinder to E' and F' respectively."
(From 1974 letter to B89.3.4 team.)

AXIS AVERAGE LINE AB AND AXIS OF ROTATION ED COINCIDE FOR A PERFECT SPINDLE. FOR A SPINDLE HAVING PERFECT CIRCULAR ELASTIC UNIFORMITY, NEITHER AXIS IS DISPLACED LATERALLY DURING ALTERATION OF THE ROTATING-LOAD CONDITIONS.

"The once per revolution radial motion of the points E' and F' will be seen as centering errors of the test cylinder but the points C' and D' will remain fixed in the axis average line as long as the weight W remains in place. Therefore, no error motion is caused by the rotating force due to the addition of weight W, but the test cylinder must be readjusted to coincide with the axis of rotation if it is to again run true."

"Now if the rotor is rotated slowly, E' and F' will describe perfect circles about the axis average line, and points C and D of the axis of rotation will be shifted to C' and D' but will remain on the axis average line."

"Consider an ideal perfect spindle. A perfect spindle must exhibit perfect circular elastic uniformity of all its structural members as well as perfect geometry of its bearing members"



"Because of the finite compliance of the bearings and other members of the structural loop, any external force or body force acting on the system will affect the relative locations of all points of the loop. Once a set of locations has been established, any alteration of the forces acting on the system will alter these locations to an extent depending on the spring rate of the bearings and the various other structural members."

EFFECTS OF THE ALTERATION OF SYNCHRONOUSLY ROTATING LOADS

AXES OF ROTATION
JULY 19 74
H. HANSEN
PROFESSIONAL INSTRUMENTS CO.

FIG 1

Figure 3.
Explanation from [1].

FIGURE 5. Dynamic response of the linear slide is used to determine the first natural frequency. The inlet air pressure was 0.55 MPa (80 PSI) for this test.

CONCLUSIONS

Displacements due to force changes are predicted by the compliance of the bearings and other structures. Assuming uniform circular elasticity, a change of force does not induce error motion, it only changes alignment. As it rotates, an off-axis load explores the stationary structure for uniform circular axial compliance. Unbalance moves the workpiece relative to the axis average line. Balance is achieved when the principal axis of inertia is on the axis average line.

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LOOKING IN VS. LOOKING OUT AT ERROR MOTION AND ECCENTRICITY

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INTRODUCTION

Standard spindle terminology refers to fixed or rotating sensitive direction but another way to describe the difference is looking in or looking out. For a lathe, we look in at the rotating structure and for a boring machine we look out at the stationary structure [1]. As for bearings, the view from outside sees imperfections of the inner races while the view from inside sees imperfection of the outer races. A rotating force from unbalance challenges the circular elastic compliance of the outer rings, the stator bearing seats, the stator, the stator support structure, and the rest of the non-rotating load path [2]. A steady force such as gravity explores for non-uniform circular elastic compliance of the inner rings, the fit of the bearing rings on the rotor shaft, and the elastic symmetry of the rotor, the chuck, and the workpiece [3]. Results vary with orientation angles respective to the stator or to the rotor.

MOTIVATION

In the past, some important basic concepts have been introduced but under-appreciated. Also, some of the implications of fixed vs. rotating sensitive direction deserve to be better-known. And, I wanted to explain it in terms even I can understand.

EXAMPLES

- An unmounted bearing illustrates the reason for duality of results [4].
- A wobble rig for boring oval shapes demonstrates how centering errors relate to oval out-of-roundness.
- Harold Arneson's example of a spindle with a notch in the rotor gives a mechanistic explanation for the relationship of these two effects.



Figure 1. A bearing has two different personalities depending on which race is examined. So you can expect to find different amounts and types of errors.



Figure 2. Turning vs. boring: The stationary tool sees rotor imperfections, the rotating tool sees stator imperfections.

The **X indicator** sees the maximum once-around because the stator support structure is soft horizontally.

The twice-per-turn signal goes unseen because the minor axis is in the non-sensitive direction.

To a fixed indicator, the signal is a once-per-revolution sine wave.

A rotating boring bar cuts ovals, while a rotating test ball looks round but with **different eccentricity** depending on the orientation angle of the displacement indicator.

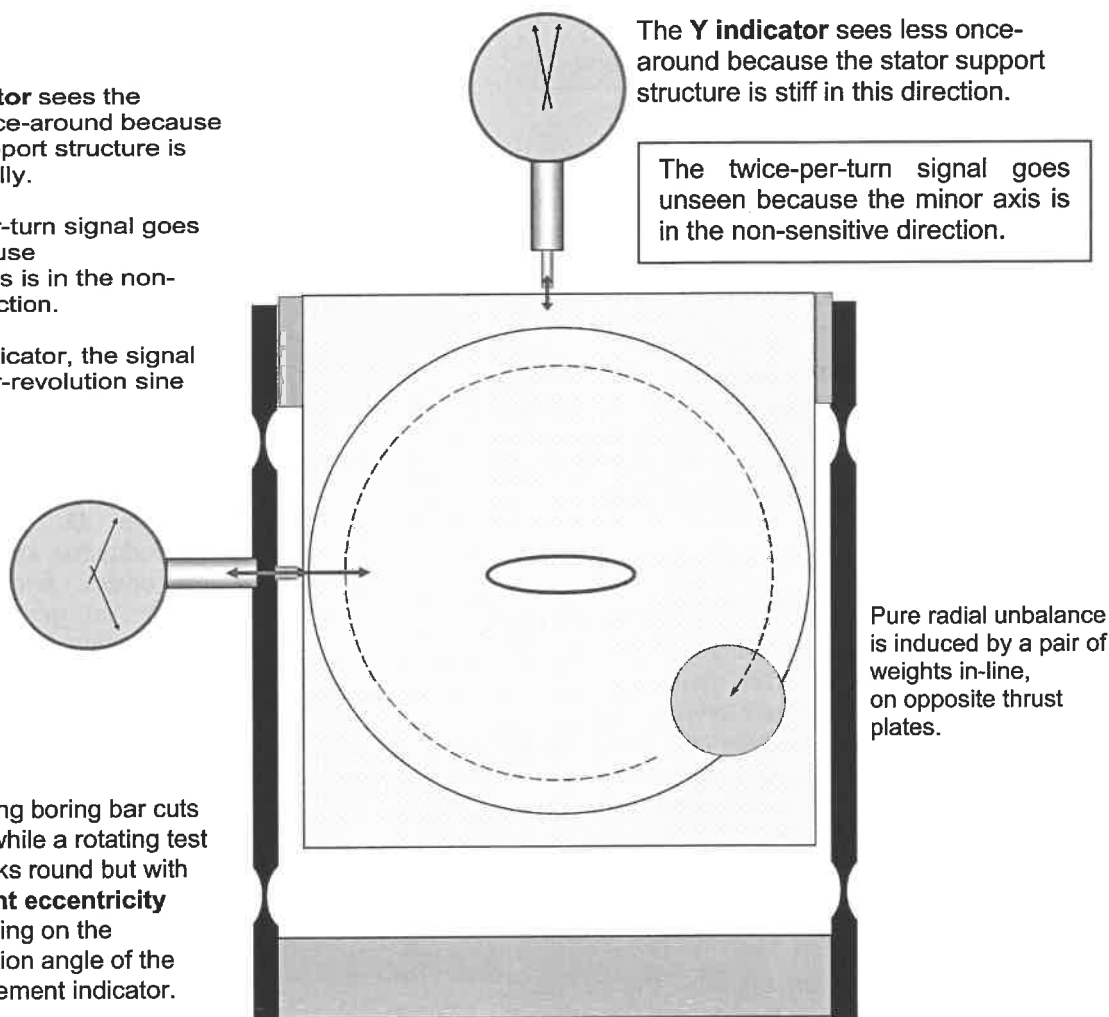


Figure 3. This wobble-bore rig is designed to generate a clean oval orbit with a lot of difference between major and minor axes and a minimum of other influences. The aim of the experiment was to illustrate how the major axis precesses as unbalance shifts from in-line with the tool point to 180° and angles in-between. The result is shown in Fig. A-20 Effect of Unbalance on Hole Boring. [1]

The **oval orbit** stays the same size, shape, and orientation as the unbalance is indexed to new angles. (What's interesting is the effect of moving the tool relative to the orientation angle of the unbalance.)

So, how can an axis of rotation cut oval shapes when the indicators show only eccentricity? The rotating boring bit is displaced in both X and Y so it copies the oval orbit. But individually the X and Y indicators see only X or Y displacement so the oval orbit is seen as major and minor amounts of once-per-revolution motion in a fixed sensitive direction.

For equations, see A-17 TWO INDICATOR MEASUREMENT FOR A FIXED SENSITIVE DIRECTION [1]. In other words, oval error motion also shows as a difference in once-around; **they are two sides of the same coin**. As a side note, this is not a spindle error motion but an axis of rotation error motion due to structural error motion of the stator supports.

Tool point oriented in-line with the bend of the rotor vs. 90° to the bend.

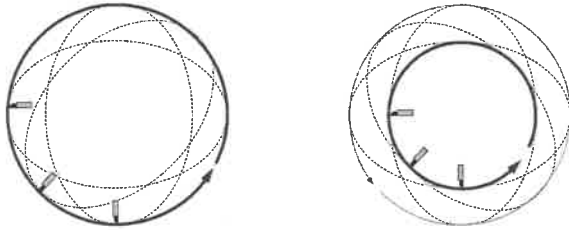


Figure 4. Boring with a bent rotor, vertical axis. Viewed from the stator, the axis of rotation moves in and out twice per turn, but viewed from the rotor the path is circular. Thus a fixed tool cuts ovals while a rotating tool cuts circles.

The circle size varies with the angular orientation of the tool point relative to the bend in the rotor. A pre-set boring bar will cut too big or too small but the hole will be round. The axis of rotation has an oval orbit – caused by the rotor – centered on the axis average line established by the stator bearings. Since the tool stays with the rotor, the tool point sweeps a constant radius and the once-around variation becomes a size error rather than a centering error. To put it another way, the view from the outside sees the bend of the rotor and the view from the inside sees a perfect stator.

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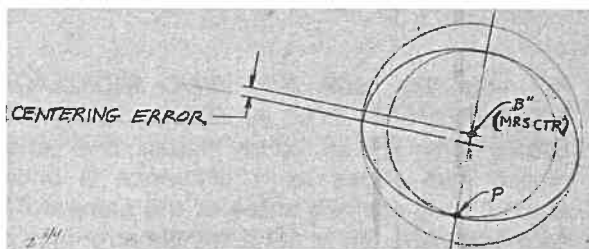


Figure 5. A different case is non-uniform bending of a spindle rotor shaft under load [2].

With the axis horizontal, gravity bends the shaft an amount that varies twice-per-revolution due to a notch, keyway, cross hole, or other discontinuity. Viewed from the outside looking in we see the variation as twice-per-revolution radial error motion. Viewed from the rotor looking out we see a circular trace and can dial

in a stationary test ball. But if the rotating indicator is indexed from the stiff to the weak orientation we see a centering error. See Fig. A-15 [1].

CONCLUSION

The presence of twice-per-revolution motion tells us to expect a like amount of variation in the once-around displacement. They are two sides of the same coin.

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Determine of Sub 50 nm CD Metrology Using by AFM

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INSTRUCTIONS

The critical dimension (CD) is decreasing rapidly in semiconductor fabrication as a result of industrial demand. In order to avoid measurement of the doublet and to support the specifications for submicron features, advanced standard laboratories in many countries are attempting to develop traceable measurement systems based on the present standards. Accurate CD determination is difficult to achieve by CD scanning electron microscopy (CD-SEM) and conventional atomic force microscopy (AFM). The specimen is prone to being burnt under the high electric current used in CD-SEM. Using conventional AFM, tip dilation is one of the problems encountered for CD measurement. NIST, used a CD-AFM instrument with a boot-shaped tip to measure the linewidth of sub 100 nm [1]. However, the high cost of this sophisticated CD-AFM system restricts its applicability. And it is necessary to calibrate the size of the boot-shaped tip in advance.

It is important to set CD standards for the linewidth, sidewall angle (SWA), line edge roughness (LER), and linewidth roughness (LWR) on a nanometer scale for further semiconductor development.

In this study, we using a stitching double-tilt image method (SDTIM) to obtain two sidewall profiles and the actual profile were obtained by images stitching technology. Measurements were carried out for various three dimensional nanoscale structures and to estimate the effect of the uncertainty factors on the CD values. Linewidth and SWA measurements were performed by DI 3100 AFM apparatus (Digital Instruments Corporation) and using the ultrasharp SSS-NCHR AFM tip (Nanosensors, Ltd.). Experimental results show that the developed SDTIM can be used for linewidth measurement with an uncertainty of less than 5 nm at a confidence level of 95%. The Best Measurement Capability (BMC) of SWA was

estimated to be 0.87° with a confidence level of 95 %. This method can also be extended to the measurement of LER and LWR. In future studies, a silicon crystal can be used as the standard for measuring the horizontal distance and vertical height to obtain an accuracy limit of 1 nm for length measurements in the sub-50 nm range.

Precision Measurement of nanoscale structure by Stitching Double-tilt Images

A scanning probe microscope (SPM) is often used for CD measurement, but the image dilation is seriously affected by the steep-side features [2]. However, the image dilation issue seriously affects on the steep-side features measured by AFM. We developed a stitching double-tilt image method (SDTIM)[3] for measuring sub- 100 nm linewidth using AFM. STDIM is based on a previously developed method for estimating uncertainty by stitching together two AFM images with different tilted angles[4].

The progress is shown in Figure 1. One scan of a linewidth sample was taken in the clockwise direction tilted and then the sample was tilted and measured again in the counterclockwise direction tilted (as shown in Figure 1-(A)). The rotation axis passed through the top surface of sample. Then, the two images were stitched using the iterative closest point (ICP) algorithm to generate an undistorted image (as shown in Figure 1-(B)). The ICP algorithm uses rotation and translation matrixes with six degrees of freedom between the relative positions of the two images. After the stitching process, the profile image (as shown in figure 1-(C)) of linewidth measurement is augmented by AFM tip. Minus the diameter of AFM tip, the image (as shown in figure 1-(D)) of linewidth is got.

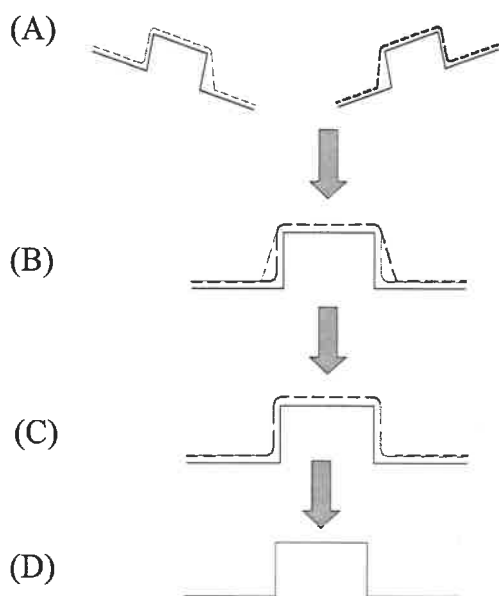


FIGURE 1. The scheme of measuring process

Precision Measurement of CD by Stitching Double-tilt Images

The experiments for linewidth measurements were performed using a DI 3100 AFM apparatus with an SSS-NCHR AFM tip. Figure 2 shows the measurement results obtained for a period pattern with a linewidth of 45 nm. The tip diameter was estimated to be 6.23 nm by using the scanning probe image processor (SPIP) software. The linewidth of measurement result after compensation is 42.2 nm. The uncertainty of linewidth measurement was evaluated as 4.4 nm with coverage factor of 2 and the major components of uncertainty is as Table 1.

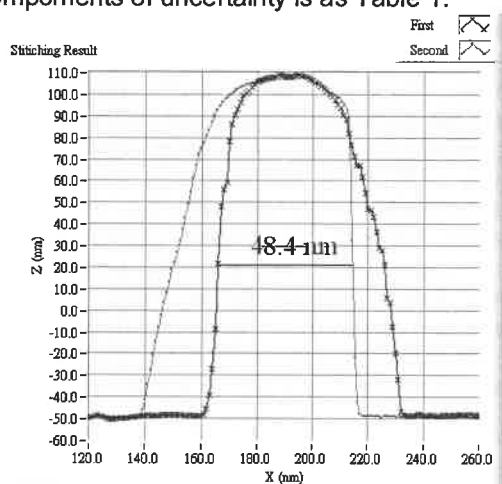


FIGURE 2. The measurement result for Linewidth

Table 1 Major components of uncertainty for sub 50 nm linewidth measurement

Source of uncertainty x_i	Value of standard uncertainty $u(x_i)$
Repeatability and traceability of linewidth measurement	0.80 nm
• Tip wear by approaching	0.13 nm
• Tip wear during measurement	0.47 nm
• Stitching method	0.65 nm
• Estimated tip radius	1.89 nm

For measurements on samples with sidewall angles approaching 90°, e.g. deep trench measurements or other semiconductor applications, there is a limitation with the depth of the trench and cone angle of AFM probe. The AFM with a tilt sample image technique is provide the sidewall angle measurement, we can get more measurement points at the vertical side of trench. As shown in Figure 3, we could obtain the equation as below:

$$d > h \cdot \tan(\alpha + \beta)$$

Where d:linewidth of trench

h: depth of the trench

α :tilted angle of the sample

β :half core angle of AFM probe

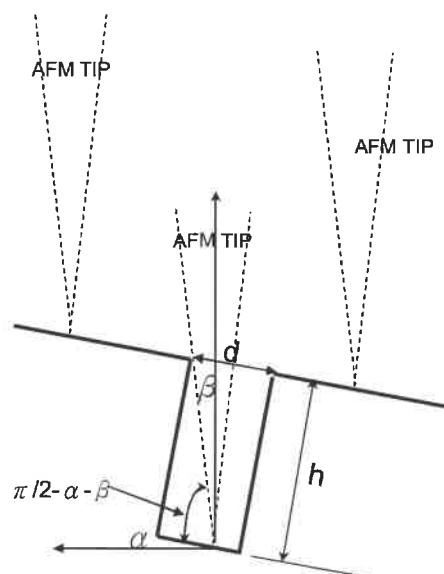


FIGURE 3 The limitation for SWA measurement

The analysis software of Labview program was used to calculate the value of SWA. The Best Measurement Capability (BMC) was estimated

Line edge roughness (LER) and Linewidth roughness (LWR) are important parts of lithography process control. The recommended LER/LWR metric is thus defined as the 3σ of residuals measured along 2- μm -long line[5] (as shown in figure 4) . This STDIM method can also be extended to the measurement of LER and LWR. Figure 5 shows the LER measurement results of 0.78 nm obtained for a period pattern with a linewidth of 45 nm. The uncertainty of LER/LWR measurement was evaluated as 1.10 nm with coverage factor of 2 and the major components of uncertainty is as Table 2.

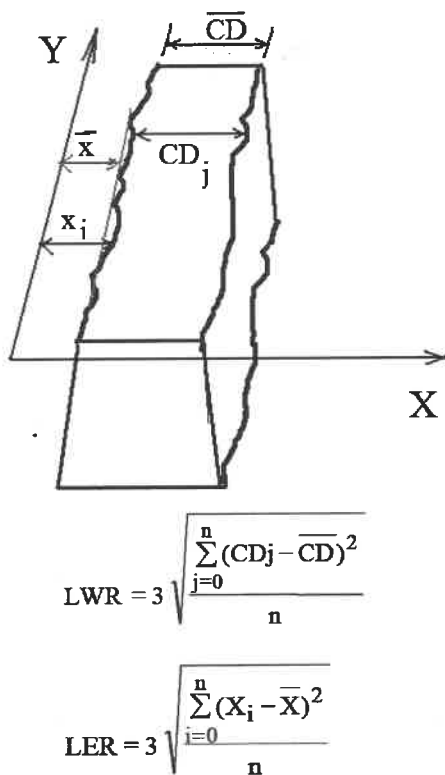


FIGURE 5. The measurement result for LER

Source of uncertainty x_i	Value of standard uncertainty $u(x_i)$
Repeatability and traceability of LER/LWR measurement	0.42 nm
• Tip wear by approaching	0.04 nm
• Tip wear during measurement	0.06 nm
• Stitching method	0.35 nm

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DESIGN OF A TEST PLATFORM TO INVESTIGATE THE ROLE OF PROCESS VIBRATIONS IN PRECISION POLISHING

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INTRODUCTION

Precision polishing using pitch tooling is still utilized in the finishing of high end optical components used in lithography, laser and other general optical applications. The process employs a high degree of operator skill which is normally accumulated over many years. High costs can be incurred during the process due to low process repeatability. Central to this issue is that the process has not received the same level of scientific investigation as other precision processes. The goal of this work is to develop a test platform to better understand how process vibrations, and significant variations of such, impact process outcomes such as material removal rates (MRR), surface finish and subsurface damage (SSD).

Pitch tooling consists of a metal platen covered with a layer of pitch. Optical quality pitch, a highly viscoelastic material, has been used for more than 300 years to generate high quality polished surfaces [1]. Until recently, the short term (millisecond domain) dynamic response of pitch had not received much attention. Previous work by this research group utilized an impact frequency response test to evaluate the effect of pitch type and age on the transient response [2, 3]. These tests demonstrated that each grade (hardness) and type of pitch (natural or synthetic) has unique dynamic properties. Further testing conducted on a bench top shaker table revealed that no pitch grade (hard or soft) provided significant vibration attenuation thus indicating that it will not attenuate any machine or process vibrations. Depending on the geometry of the pitch samples, it was found that they could resonate at frequencies over 10 kHz resulting in significant vibration amplification [4]. Passive damping materials have been identified to provide attenuation of all vibrations above 600 Hz [5]. The passive damping material (such as cork) is adhered to the metal platen before the pitch is poured.

During polishing the workpiece is expected to experience some level of steady state process vibrations. Previous work by this research group measured process vibrations on two different polishers operating under identical conditions with the same polishing tool transferred between polishers for testing [6]. The magnitude of the process vibrations and their frequency content varied from machine to machine, and with process parameters such as platen speed and over-arm swing rates. Figure 1 illustrates the root mean square (RMS) of the vibration response between 10 Hz to 16 kHz. The amplitudes of process vibration of the New machine is in the order of 10 nm at 250 Hz, 5 nm at 500 Hz and become negligible above 1.2 kHz (<0.5 nm). These vibration amplitudes are much lower than those recorded in vibration assisted polishing process (frequency ≥ 20 kHz), where the amplitude varies from 2 μm to 40 μm [7, 8]. Significant differences between the Old and New machines are also evident. The vibration of the Old machine varies by approximately 65% depending on operating parameters. Converting into the frequency domain producing Fast Fourier Transform (FFT) plots, provides the frequency content and the corresponding amplitude of the vibrations. Figure 2 provides the corresponding FFT of both the machines operating with a platen speed of 20 rpm. The FFTs indicate that at higher frequencies the magnitudes of the vibrations present on the New machine (while low) are greater than on the Old machine (negligible vibrations). Work currently underway suggests that the different vibration signatures do impact the process outcomes.

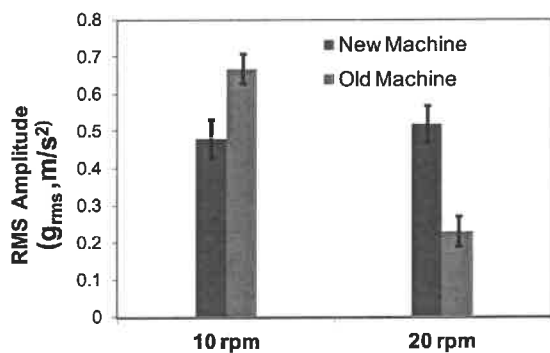


FIGURE 1. RMS vibration amplitude on both machines ($g = 9.81 \text{ m/s}^2$).

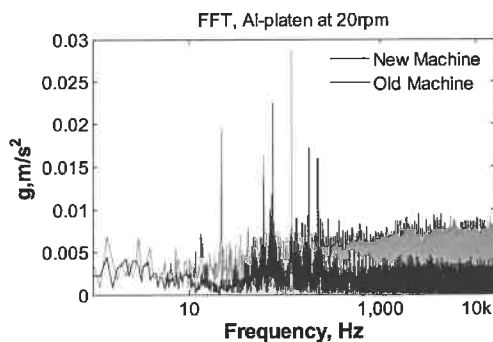


FIGURE 2. FFT of vibrations when the platen is rotating at 20 rpm.

To get at the fundamentals of the material removal mechanism it is necessary to isolate vibrations of different frequencies and amplitudes and directly measure their impact on the process outcomes. As such a task would be difficult on an actual polisher; a test-platform was specifically designed, fabricated and characterized. The following sections detail the platform design, vibration characterization, and the test plans.

PLATFORM DESIGN

The set-up has two main components, a shaker table to excite the pitch tool, and a translation stage to provide relative motion between the sample and the tooling, see figure 3. The stage is driven by a low friction, low vibration precision ball screw mechanism. Two chrome-plated precision shafts and four self-aligning linear bearings support the stage. The translation distance is controlled by two SPDT push button switches on either side of the stage. When the stage hits the switch the mechanical relay circuit changes the direction of the motor rotation

reversing the stage direction. The stage is driven by a 12V DC motor with a maximum speed of 190 rpm. Power is supplied to the motor by a BK PRECISION DC voltage generator. The maximum travel of the stage is 300 mm with a maximum velocity of 15 mm/sec.

The workpiece holder consists of an aluminum block which is connected to the horizontal moving stage by two precision shafts, via linear bearings. This will allow smooth motion in the vertical direction while preventing sample tilt and tip as it moves across the pitch tool. The weight of the aluminum block (≈ 500 grams) is selected such that the applied load on the workpiece is representative of polishing loads. Additional mass can be added if needed. A vinyl gasket between the workpiece and the holder is used to help prevent higher frequency vibrations being transmitted to the workpiece from the stage.

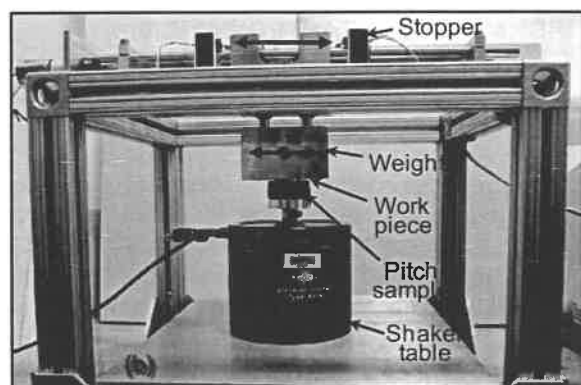
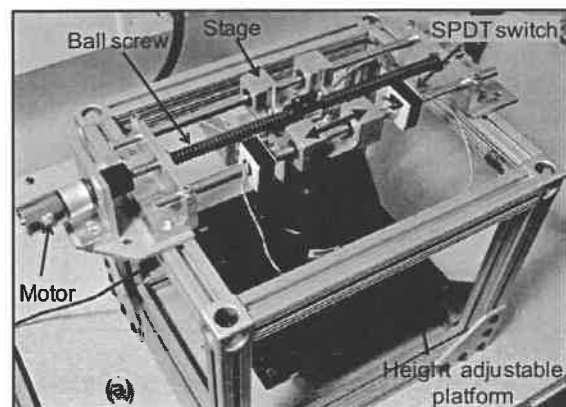


FIGURE 3. (a) Top and (b) side view of the test platform.

The pitch tool, a 22 mm thick layer of pitch on a 50 mm diameter aluminum platen, is screwed onto a shaker table (BK Vibration Exciter Type

4809). The shaker table which is connected to a frequency generator and a vibration amplifier can vibrate up to 20 kHz with a maximum displacement of 8 mm. The shaker table has resonance at 20 kHz. A dynamic load sensor is placed between the pitch sample and the shaker table to observe the forces during a polishing test. A schematic of the test set-up is given in figure 4. Polishing slurry is supplied to the polishing region by a separate delivery system which is not shown in the schematic.

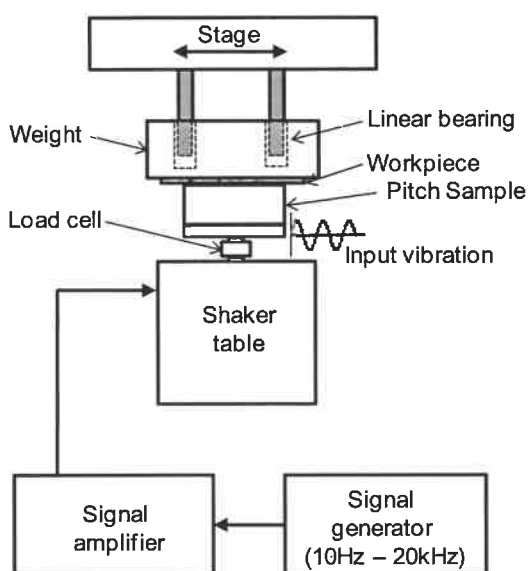


FIGURE 4. Schematic of test operation.

VIBRATION CHARACTERIZATION

The vibrations in the platform due to the stage movement are characterized for different running conditions. A PCB accelerometer (352B10) that operates up to 17 kHz is used to measure the vibration. The accelerometer attached to the workpiece is connected to a computer through a signal conditioner, an ADC converter, and a National Instruments DAQ card (6036E), which has a maximum sampling rate up to 200 kS/s. LabView is used to record the accelerometer's response and to convert from the time into the frequency domain producing a FFT plot. Ideally the vibration magnitude generated by the translation stage must be significantly less than that generated by a typical polishing process. An initial goal was set at 80% less than that produced by the process. To assess the level of vibrations input into the workpiece by the mechanical translation of a

glass workpiece over a pitch tool the vibrations were measured over 5 seconds while the stage ran at 0.5 m/min. The shaker table was not vibrating. Figure 5 shows the FFT of the vibrations. The 60, 120 and 180 Hz spikes represent the system power supply frequencies and work is underway to remove. Figure 6 compares the results to the vibration signatures obtained from the Old and New machines. At lower frequencies (10 Hz to <250 Hz) the amplitude of the vibrations are similar for all systems, however at frequencies above 251 Hz the platform and the old machine have comparable signatures. These amplitudes are less than 20% of that generated by the new machine. The platform could be used in a conventional polishing set up to translate the part over a tool.

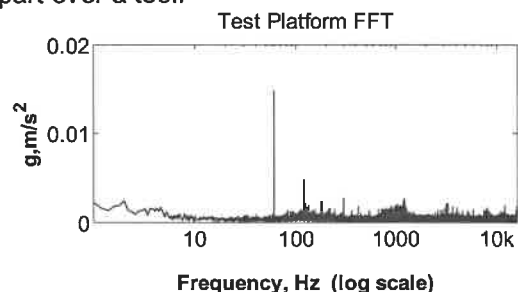


FIGURE 5. Platform vibration with no shaker table excitation.

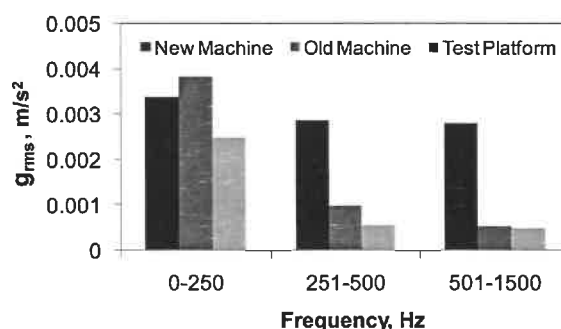


FIGURE 6. RMS vibration amplitude at different frequency bandwidths.

As stated in the introduction the calculated peak displacement magnitudes for the New machine at 250 Hz, 500Hz and 1.2 kHz are 10 nm, 5 nm and 0.5 nm respectively. Achieving these low levels of vibrations is challenging for any mechanical system. Of interest is to determine the best performance of the system at set frequency levels. To achieve this, the magnitude of the vibrations experienced by the workpiece

as it translates over the vibrating pitch tool was recorded at tool frequencies of 250 Hz, 500 Hz, 1 kHz and 10 kHz. Figure 7 illustrates the accelerometer output over 0.04 s when the tool was vibrating at 250 Hz. This level of vibration corresponds to a peak displacement of 500 nm. Figure 7 was obtained with the lowest shaker table drive voltage that produced a discernible sine wave. The peak displacements at 500 Hz and 1 kHz were 90 nm and 20 nm respectively. At 10 kHz displacements are in the order of 1 nm. As these values are all higher than the actual polishing process the test platform will not be able to directly replicate the polishing process. That said the associated accelerations achieved with the test platform are still many orders of magnitude lower than those used in vibration assisted polishing processes and thus the test platform will enable an investigation into what affects the process outcomes more; amplitude of vibration, or frequency of vibration.

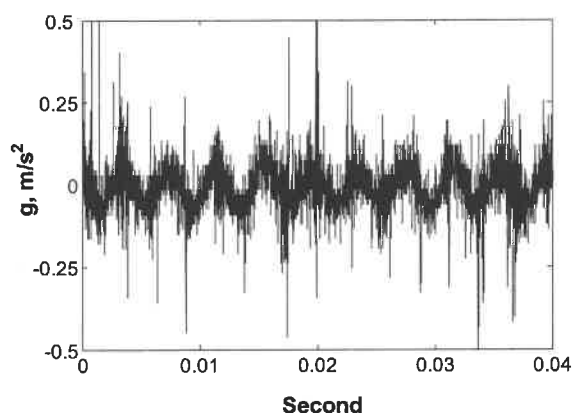


FIGURE 7. *Vibrations measured on the workpiece as it translated over a vibrating pitch tool (250 Hz).*

SUMMARY AND FUTURE WORK

A test platform consisting of a translation stage and a shaker table has been built and characterized to separate the impact of vibration frequency and amplitude on precision polishing outcomes such as material removal rates, surface finish and SSD. Initial tests will focus on polishing fused silica samples with 1 micron ceria.

ACKNOWLEDGEMENTS

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A SHORT-RANGE ERROR EVALUATION OF A WAVELENGTH MODULATION ENCODER

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INTRODUCTION

Demands for accuracies of linear scales are constantly increasing in precision engineering field, such as ultra-precision diamond cutting machines and semiconductor lithography tools. Usually, these tools use a compensation table to compensate low spatial frequency errors of linear encoders. However, high-spatial-frequency errors, or in other words short-range errors, cannot be compensated using the compensation tables. Therefore, the short-range error is a more serious concern for today's precision industry, and these errors are considered as one of the primary performance indices of linear encoder productions.

Last year we introduced a laser wavelength modulation encoder which has an ultra high digital resolution, 3.8pm [1]. An advantage of the encoder is the low interpolation error and an ease of handling by users by using practically large pitch scale. In this paper we present two short-range-error evaluations of the wavelength modulation encoder.

First is the interpolation error evaluation across the entire gap and angle tolerance between the head and the scale. Second is the evaluation of the scale's short-range errors. The errors are often caused by stitching exposures during the scale manufacturing. However, the wavelength modulation encoder uses a 4 μ m-pitch reflective grating which is manufactured using high-precision lithography tools. These machines minimize the scale stitching errors. This paper also reports the accuracy in the short-range.

THE CONCEPT OF THE WAVELENGTH MODULATION ENCODER

Figure 1 illustrates the concept of the wavelength modulation encoder. A sinusoidal-modulated current drives the laser diode, and the laser wavelength changes with the current.

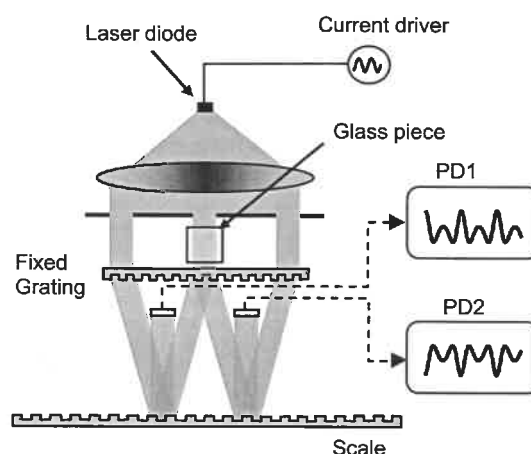


FIGURE 1. The concept of the wavelength modulated encoder.

The beam out of the laser diode is first collimated by the collimator lens, then goes through an aperture separating the beam paths into three beams. The middle beam goes through a piece of glass to retard the phase compared to the outside beams. Then the middle beam and the outside beams go through a fixed grating. Diffraction beams are generated when each beam goes through the fixed grating. As shown in Figure 1, the +1 and -1 order diffraction beams diffract again on the scale which is placed on the moving stage. The fixed grating and the scale have the same pitch; therefore, interference signals are detected on the photo diode at PD1 and PD2 positions.

The amount of phase delay caused by the glass is a function of the wavelength, so the phase of the diffraction light coming into the scale shifts as a function of the wavelength. The light diffracted by the scale will experience a 2π phase shift when the scale grating moves half of

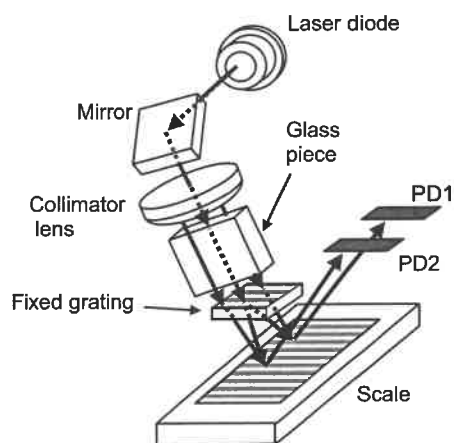


FIGURE 2. Production model optics.



FIGURE 3. Optical head and scale.

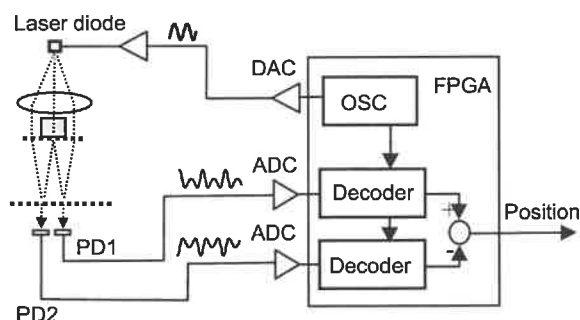


FIGURE 4. System configuration/

its period. Then, modulation of the laser diode wavelength will result in a sinusoidal phase shift generating many harmonic signals which includes the scale position information. Figure 2 shows the actual optical design inside the encoder head. To avoid undesired optical

feedback from the scale surface, the laser diode is located in an offset position. The laser beam is reflected by a mirror, before passing through a collimator lens. The optical path length of the glass piece is 5 mm, and its refractive index is 1.4. Both the fixed and the scale grating are 4 μ m pitch.

Figure 3 is a photo of the optical head and the scale. The head unit is packaged within a 32 mm x 20 mm x 18 mm enclosure. The scale has 4 μ m pitch hard-chrome-coated phase grating. This scale is easy to handle, with a good performance against the environment.

Figure 4 illustrates the system configuration of the encoder system. A modulation signal oscillator (OSC) and two decoders are provided in single FPGA. The PD1 and PD2 signals are decoded into position information separately. The laser wavelength drift can be canceled and the positions averaged by subtracting PD1's position output from PD2's.

RESOLUTION AND POSITION STABILITY

Before the short-range accuracy evaluation, we must check the encoder output stability in static position. Figure 5 shows the position output stability of the encoder at room-temperature. Figure 5(a) is a ten-second-long time-trace data at 10 kHz sampling. The standard deviation (STD) of the whole data set is 23.1 pm. Figure 5(b) is the Allan standard deviation, which shows the signal deviation versus averaging time. The Allan standard deviation (or Allan variance) is widely used for stability evaluations of gyroscopes and frequency oscillators. The data reveals the stability of the encoder at low frequency is around 2 Pico-meter which is sufficient for any short-range accuracy evaluations which is around 1-nm or less.

INTERPOLATION ERROR

A low interpolation error is the most important advantage of the modulation encoder. The encoder uses single waveform instead of two sinusoidal signals in conventional encoders; therefore, there is no offset or amplitude errors of the signals. However, unexpected optical signal distortion causes an interpolation error. The signal distortion is likely to occur when the encoder head is misaligned. Figure 6 shows the interpolation error versus the gap and angles between the optical head and the scale. Tolerances in the figures are the allowable working ranges on customer's machine; gap (± 0.2 mm), yawing (± 0.52 mrad), pitching (± 1.3 mrad), and rolling (± 1.0 mrad). The interpolation

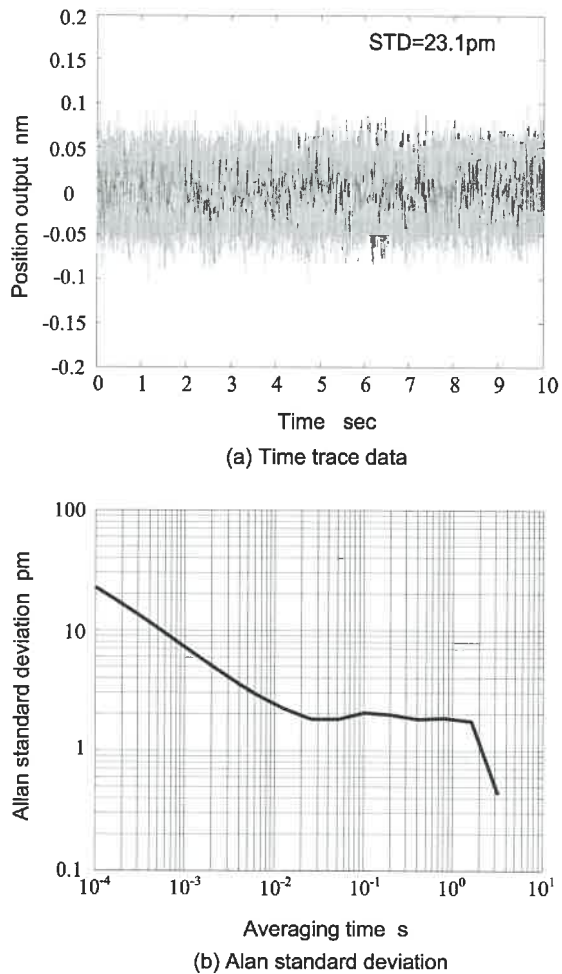


FIGURE 5. Position output stability.

error keeps within under 1nm-pp in whole tolerance ranges.

THE SCALE SHORT RANGE ERROR

Figure 7 illustrates the concept of the scale-short-range error measurement system. The two encoder heads are fixed at 15mm-apart along the scale. The difference between the two output signals during the scan indicates the scale linearity change in a 15mm segment. If the distance between the heads is steady enough, the short-range-error can be measured accurately. However, it is necessary to note that the peak-to-peak value of the difference signal appearing at twice the scale error as shown.

Figure 8 shows results of the small-range-error measurement of a quartz phase grating scale manufactured by Nikon. Figure 8(a) is the raw data, (b) is the data after subtracting the moving average over 1mm in order to extract the high

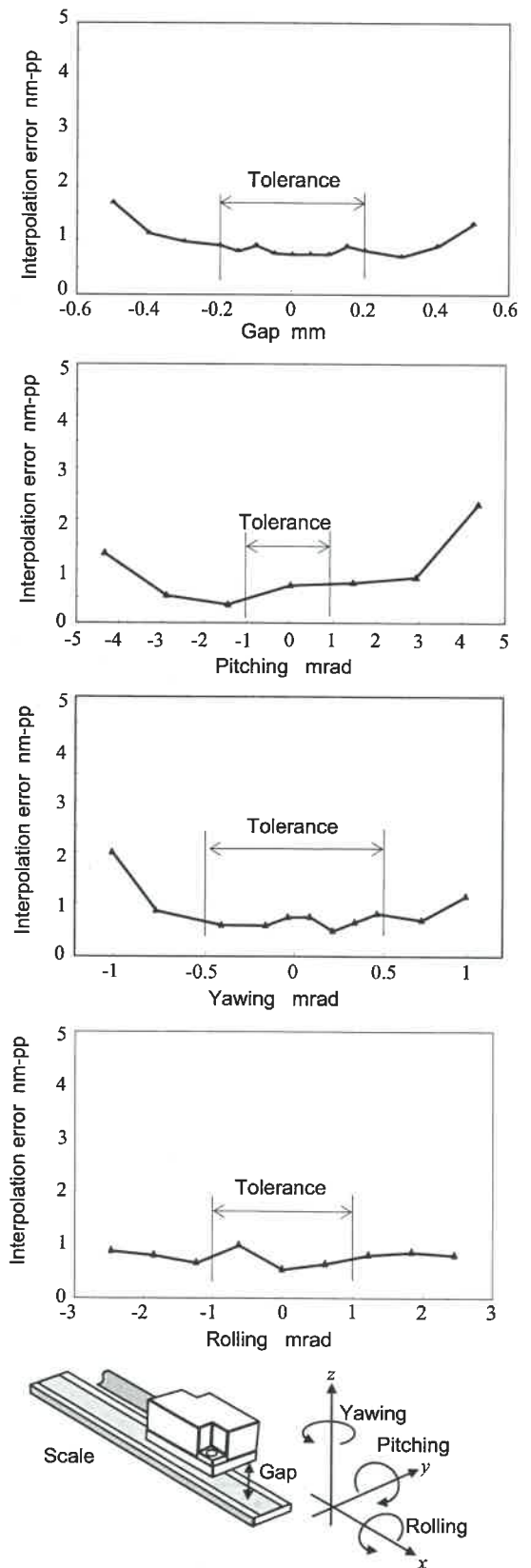


FIGURE 6. Interpolation errors vs. gap & angles.

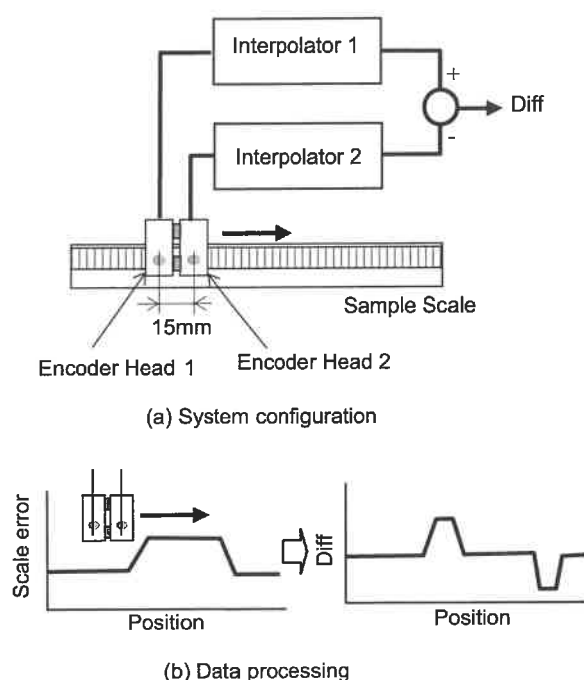


FIGURE 7. Short-range-error measurement.

frequency components, and (c) is the zoomed data of (b). The graphs in Figure 8 show 6-time overlapping scan data. The repeatability of the measurement was 0.088nm sigma in each data point. It shows that the short-range-error measurement with this system is very reliable. Finally, we concluded the maximum short-range-error in any 1mm segment of the scale is around 1nm-pp.

CONCLUSIONS

This study reveals the short-range accuracy of the modulated laser encoder. The two major components of the error are the interpolation-error and the short-range error of the scale. The wavelength modulation encoder provides a low interpolation error due to the decoder technology. We obtained within 1nm-pp interpolation error across the wide gap and large angle tolerances. For the second evaluation, we introduced a short-range-error measurement system using two encoder heads. This machine provides high repeatability due to the encoder output stability. The evaluation result of a quartz phase grating scale was within 1nm-pp short-range-error in whole scale length, by using this measurement system.

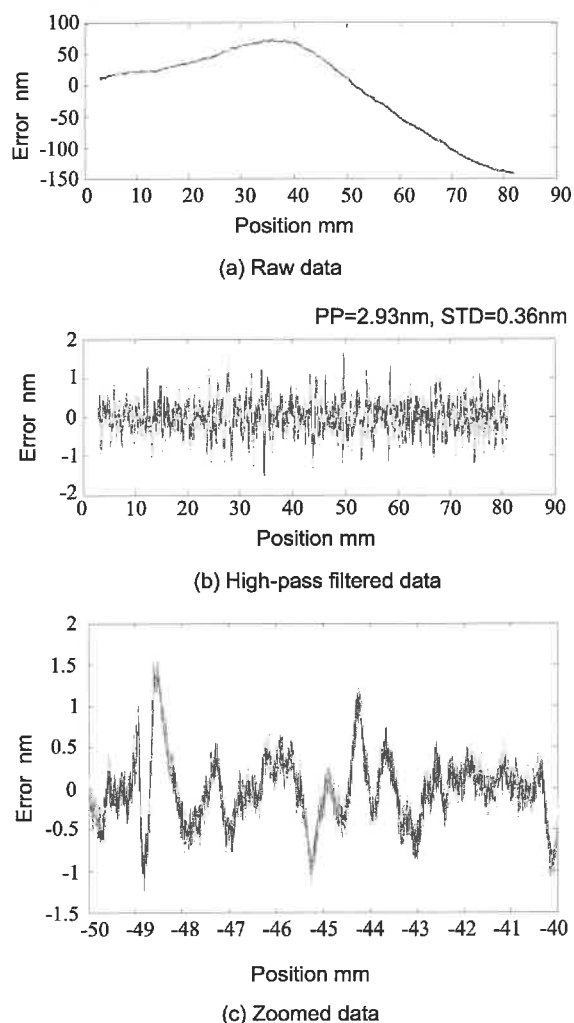


FIGURE 8. Scale-short-range-error data

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EVALUATION OF PROPOSED TEST ARTIFACTS FOR FIVE-AXIS MACHINE TOOLS

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INTRODUCTION

In machine tool metrology, there are two primary ways to evaluate the performance of a machine tool: through a series of instrumented tests and through the manufacturing of test pieces. Advantages of test pieces are that machining parts is more akin to the actual purpose of the machine tool and that they do not require specialized measuring instruments. The disadvantages are that test pieces are composite tests—all errors present in the machine tool contribute to errors in the part—and part production is complicated by cutting forces and tool dynamics.

An early draft of an international standard [1] includes the introduction of two artifacts to test coordinated motion of 5-axis machine tools. The current work evaluates the production of these artifacts by discussing lessons learned and comparing axis trajectories and measurement results for each artifact. When possible, results are compared to analogous instrumented tests.

ARTIFACT DESCRIPTIONS

The most simple design that sufficiently tests the machine tool is desired to ease measurement, analysis, and correlation of part errors to machine error motions. A frustum (truncated cone) is the simplest geometry that can be produced using simultaneous 5-axis motion. The frustum artifact is already known to the machine tool community [2-5]. The frustum manufactured in this study is shown in Figure 1(a). The size differs from sizes prescribed in the draft standard because the current work was initiated before the release of the draft.

The second artifact is a truncated square pyramid (TSP), shown in Figure 1(b). This artifact is newer to the machine tool community.

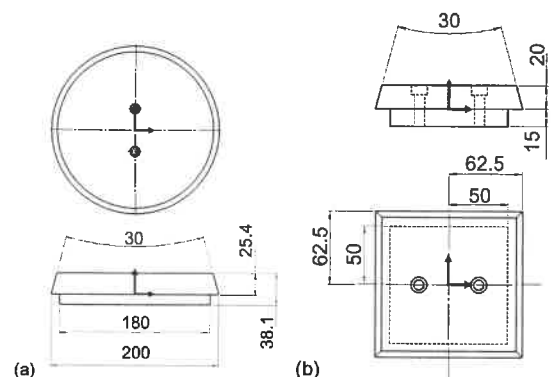


FIGURE 1. Engineering drawings of artifacts investigated in this study: (a) cone frustum, (b) truncated square pyramid. Length units are mm, angle units are degrees.

MACHINE

The machine used for the study is configured with a tilting-rotary table. The machine has a non-orthogonal B'-axis that is inclined from the Y-axis by 45°. The travel ranges are 800 mm for the X-axis, 700 mm for the Y-axis, and 550 mm for the Z-axis. The B'-axis ranges from 0° to 180° and the C'-axis has infinite rotation.

TOOL PATH

We chose similar cutting conditions to those prescribed by the draft standard: speed = 300 m/min, feedrate = 0.05 millimeters per tooth, tool = 12.7 mm diameter carbide cutter for Aluminum. The finish cut is made by side milling with a 0.1 mm radial depth of cut.

¹ Official contribution of the National Institute of Standards and Technology (NIST); not subject to copyright in the United States. The full descriptions of the procedures used in the paper require the identification of certain commercial products. The inclusion of such information should in no way be construed as indicating that such products are endorsed by NIST or are recommended by NIST that they are necessarily the best materials, instruments, software or suppliers for the purposes described.

Two stipulations must be made to ensure that all five axes are in simultaneous motion while machining. First, the artifacts are inclined about the Y-axis. Two frustum inclination angles are tested in this study: 10° and 15°. A 20° inclination angle is used for the TSP. Second, for the TSP, the tool path must be such that the vector describing the tool centerline must continually intersect the pyramid axis.

The tool paths were generated by commercially available computer aided manufacturing software. The machine controller allows programming using tool orientation vectors for the rotary axes and work coordinate system (WCS) positions for the linear axes.

SETUP

The frustum is mounted with the WCS origin offset from the C'-axis by a distance R , 50 mm in this study. The TSP is mounted such that the WCS origin lies on the C'-axis.

The machine tool utilized in this study is equipped with a touch trigger probe that is typically used to establish the position of the WCS origin and the orientation of the WCS with respect to the machine coordinate system. However, this proved problematic for these artifacts. Because the artifacts' work surfaces are not parallel to machine axes or WCS axes, the canned WCS cycles in the machine's controller could not be used directly on the artifacts. Custom fixtures were designed that had surfaces parallel to the machine axes, but uncertainty in mounting the artifact meant the part could not be cut with only one finish pass. User-written probing programs that contacted the actual work surfaces were written, but again excessive uncertainty prevented machining in one finish pass. These difficulties led to the decision to start with a part blank that was significantly oversized (5 mm), coarsely position the part in the desired programmed position, and use roughing passes before the final finish pass. The initial roughing pass may have variable depth of cut, but it trues the part for the finish pass.

AXIS TRAJECTORIES

The machine program was run through a computer model simulating the machine tool controller to generate the individual axis trajectories. The combination of the part's opening angle and the setup inclination angle determine the trajectories of the rotary axes.

The trajectories of the linear axes are influenced not only by these same angles, but also by the position of the part within the machine's work volume (*i.e.*, R). The axis trajectories for the setups explored in this study are shown in Figures 2, 3, and 4.

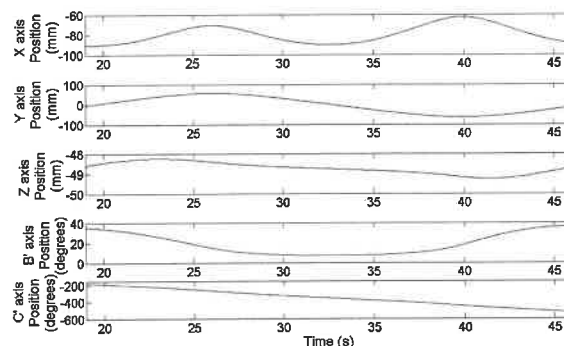


FIGURE 2. Axis trajectories for frustum with 10° inclination angle and setup offset from the C'-axis by 50 mm in the x-direction.

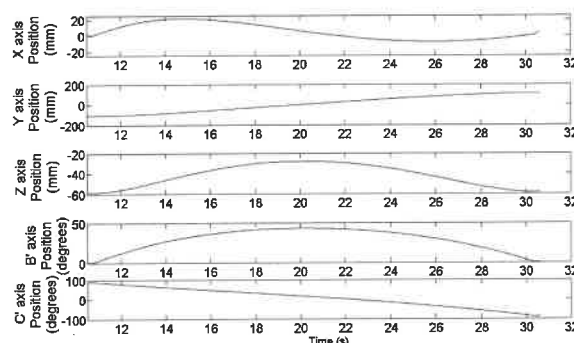


FIGURE 3. Axis trajectories for frustum with 15° inclination angle and setup offset from the C'-axis by 50 mm in the x-direction.

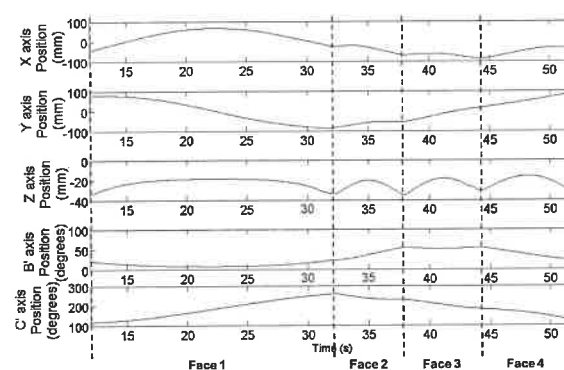


FIGURE 4. Axis trajectories for the TSP inclined by 20° and setup on the C'-axis.

The individual axis trajectories for the two frustum tests are very different (see Figures 2 and 3). For the 15° inclination angle, the B'-axis makes a relatively smooth transition from 0° to

$\approx 43^\circ$ and back to 0° to achieve the cone angle. The C'-axis rotates 180° at a near-constant velocity to trace the frustum surface. For the 10° inclination angle, the B'-axis ranges from $\approx 7^\circ$ to $\approx 36^\circ$ and back, but the C'-axis makes a full 360° rotation (from $\approx -193^\circ$ to $\approx -553^\circ$). The trajectories of the linear axes are also quite different, but the ranges of these travels could be adjusted by selecting different positions for the test pieces within the work volume.

Also of note is that the axis trajectories for the 15° frustum test are different than the trajectories used to cut the same part with an orthogonal rotary axis configuration [3,4]. Specifically, on a machine with an orthogonal tilting-rotary table, the C'-axis travels 360° .

The trajectories for the TSP appear much more complex than those for the frustum tests. The trajectories for each face vary in range of motion and time. In fact, the trajectories for each face of the TSP can be thought of as four separate coordinated motion tests.

RESULTS

The draft standard prescribes what measurements should be taken, not the instruments that should be used to take them. As such, several possibilities for measurement are shown. For the frustum, the only measurement prescribed is the roundness of the machined surface. For the TSP, the straightness of each face is measured along with the perpendicularity of two faces to one reference face and parallelism of one face to the same reference face. Each part was measured at approximately its mid-height.

The frustum created using a 10° inclination angle was measured by a coordinate measuring machine (CMM) in two different ways: by contacting the work surface at 36 individual points and by scanning the CMM probe along the work surface. The results of the point-by-point measurement exhibit a non-circularity of $7\text{ }\mu\text{m}$. The scanning measurement, shown in Figure 5a, appears slightly different. A notch or divot at 0° (the point of entry/exit for the tool) is quite apparent. Also, we noticed CMM probe hunting at the start of the measurement (the asterisk in Figure 5a). As such, the deviations immediately following the start of the measurement result from the measurement process, not from the artifact. If the notch and the hunting sections are ignored, the roundness

is $6\text{ }\mu\text{m}$. The expanded uncertainty in the CMM measurement is $3.4\text{ }\mu\text{m}$ over a travel of 450 mm .

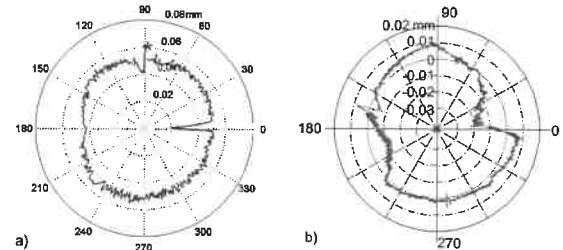


FIGURE 5. Results of measuring: a) 10° frustum test by scanning the CMM probe, and b) 15° frustum test by roundness tester.

The frustum created using the 15° inclination angle was measured on a roundness measuring machine with sub-micrometer expanded uncertainty. Figure 5b shows the results of this measurement, which reveals a different error profile. Again at the tool entry/exit point (*i.e.*, 0°), one can see a step of approximately $32\text{ }\mu\text{m}$. The deviations on the opposite side of the part (from approximately 165° to 225°) are likely the result of chatter during the cutting process. Evidence of chatter was detected audibly during cutting and visibly on the part's surface.

The TSP was measured by scanning the CMM probe. The error profiles of each face, along with the best fit lines to those profiles, are shown in Figure 6. The straightness values for face 1 (facing toward the $-X$ at setup), face 2 (facing toward $+Y$ at setup), face 3 (facing toward $+X$ at setup) and face 4 (facing toward $-Y$ at setup) are $18\text{ }\mu\text{m}$, $13\text{ }\mu\text{m}$, $15\text{ }\mu\text{m}$, and $7\text{ }\mu\text{m}$, respectively. Face 1 was taken as the reference surface. The perpendicularity of faces 2 and 4 to face 1 are $136\text{ }\mu\text{m}$ and $139\text{ }\mu\text{m}$, respectively, and the parallelism of face 3 to face 1 is $134\text{ }\mu\text{m}$.

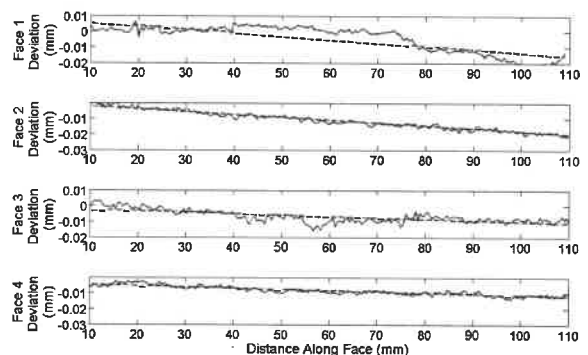


FIGURE 6. Results of measuring the TSP faces by scanning probe CMM.

BALLBAR TESTS

The TSP has no analogous instrumented test, but the frustum tool path can be checked using a telescoping ballbar. In fact, a draft international standard prescribes this as one of the tests for 5-axis coordinated motion [6]. The results of these tests are shown in Figures 7a and 7b for the frustum with the 10° inclination angle and the 15° inclination angle, respectively.

The profile in Figure 7b shows a step at the start/stop point similar to the one seen in Figure 5b, confirming that this step is indeed an error motion of the machine tool. The profile in Figure 7a shows a non-circularity of 6 μm , similar to the roundness observed in point-by-point CMM measurement, but the center of the entire ballbar profile is shifted by approximately 12 μm . This shift in the center is indicative of a location error in the rotary axis and any residual setup error after properly centering the ballbar spheres [7]. Also, the lack of a divot at the start/stop point indicates that this deviation in the artifact results from the cutting process, not from an error motion of the machine tool.

While the 10° and 15° results appear much different, closer examination reveals the similarity. The consistency in the plots appears if one compares the entire 15° ballbar plot with half of the 10° plot. If one takes the start point of the 10° test at 90° and the finish point at 270°, one notices a step of approximately 24 μm , very similar to the 25 μm step in the 15° plot. This comparison is valid because these are the portions of the plots over which the C'-axis positions are approximately equal to each other during cutting. In fact, previous measurements on this machine indicate a location error of the C'-axis in the machine's X-direction of approximately 10 μm [7].

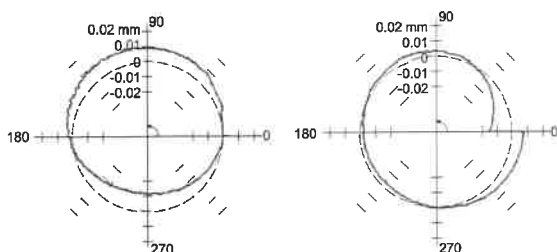


FIGURE 7. Results of measuring tool path deviation with a ballbar, a) in the 10° frustum test, and b) in the 15° frustum test. Expanded uncertainty ($k=2$) in ballbar measurement is 0.001 mm.

CONCLUSIONS

When machining test artifacts that do not offer paraxial surfaces, roughing passes may be necessary. When performing these roughing passes, one should take care to avoid unstable cutting conditions because chatter during roughing may be replicated in the finish pass.

The relative simplicity of the frustum and its associated axis trajectories along with an analogous instrumented test make it the preferred artifact to a truncated square pyramid. This simplicity is evidenced by the ability to link the observed test part error profile with a location error in the machine's C'-axis.

The cone frustum was tested at two different inclination angles that showed significantly different results. The result of the 10° frustum test is a deviation of 7 μm , while the result of the 15° frustum test on the same machine (with the same errors) is 32 μm . As such, care should be taken when judging the performance of a machine tool by machining only one artifact at one setup configuration.

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Development of an Optical Measurement System for Five-Degrees-of-Freedom Error Motions of a Micro High-Speed Spindle

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INTRODUCTION

The miniaturization of machine tools and cutting tools in recent years has led to an increasing need for spindles designed for high-speed rotation and reduced spindle size. The evaluation of spindle error motions is essential in order to examine the performance of spindles and reveal points for improvement. In addition, the form accuracy and surface roughness of workpieces are extremely dependent on the rotational accuracy of the microspindle, and this development has brought about an increasing demand to evaluate microspindle rotation errors.

It is very difficult to evaluate five-degrees-of-freedom error motions (one axial, two radial, and two angular motions) for high-speed microspindles using the conventional method, which simultaneously employs a master ball or cylinder and displacement sensors (e.g., an electrostatic capacitance displacement meter). This difficulty is due to limitations in the measurement range, mechanical interaction, and frequency-response characteristics. Because the reference pieces used in the conventional method, such as the master ball or cylinder, must be large enough to allow the displacement sensors to measure them, they are not appropriate for microspindles. High-speed microspindles can only accommodate small, lightweight reference pieces. Therefore, it is very difficult to measure spindle-rotation errors for microspindles. Furthermore, the conventional system is very expensive because five displacement sensors are necessary in order to measure the five-degrees-of-freedom error motions.

In recent years, some optical techniques to measure high-speed microspindles have been reported [1, 2], but these methods cannot be applied to a simultaneous measurement of five-

degrees-of-freedom errors. We present herein a simple and low-cost measurement system to measure simultaneously five-degrees-of-freedom error motions of high-speed microspindles. The prototype of the measurement system has been fabricated and evaluated [3]. In this study, measurement error caused by roundness of lens shape is analyzed using ray tracing and the measurement system is experimentally evaluated.

MEASUREMENT PRINCIPLE

Figure 1 shows an illustration of the measurement system of five-degrees-of-freedom error motions for a micro high-speed spindle. This measurement system is composed of a rod lens, a ball lens, four laser beams (L1, L2, L3, L4), four condenser lens (CL1, CL2, CL3, CL4), and multiple divided photodiodes (PD1, PD2, PD3, PD4). The ball lens of 3 mm in diameter is affixed at the end of the rod lens of 3 mm in diameter, which is mounted on the chuck of the spindle. The stem of the rod lens is irradiated with a laser beam emitted by a laser

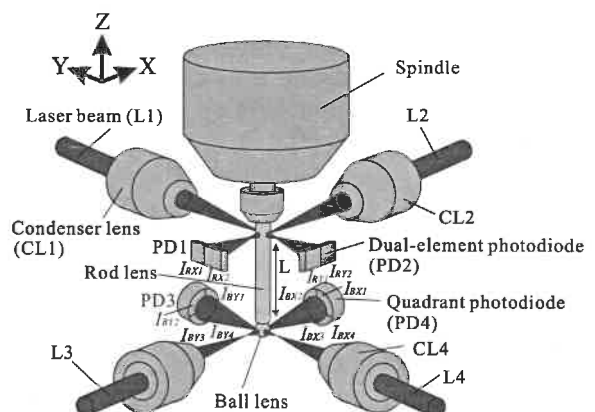


FIGURE 1. Optical measurement system.

diode in the X and Y directions. The two dual-element photodiodes (PD1, PD2) are opposite to the two condenser lens (CL1, CL2), and the rod lens is installed between the condenser lens and the dual-element photodiode in an orthogonal position. The laser beams that penetrate the rod lens reach the two dual-element photodiodes. The intensities, detected by the two dual-element photodiodes, are converted into voltage and are defined as I_{RX1} , I_{RX2} , I_{RY1} , and I_{RY2} , as shown in Fig. 1. Similarly, the ball lens is irradiated with a laser beam emitted by the laser diode in the X and Y directions. The quadrant photodiodes (PD3, PD4) are opposite to the condenser lens (CL3, CL4), and the ball lens is installed between the condenser lens and the quadrant photodiode in an orthogonal position. The laser beams that penetrate the ball lens are received by the two quadrant photodiodes. The intensities, detected by the two quadrant photodiodes, are converted into voltage and are defined as I_{BX1} , I_{BX2} , I_{BX3} , I_{BX4} , I_{BY1} , I_{BY2} , I_{BY3} , and I_{BY4} , as shown in Fig. 1.

Figure 2 (a) also shows a cross section of the rod lens irradiated by the laser beams in the XY plane. When the spindle rotates without rotation errors, the laser intensities measured by each element of each photodiode are equal ($I_{RX1} = I_{RX2}$, $I_{RY1} = I_{RY2}$). When the spindle rotates with rotation errors (Figure 3 defines the five-degrees-of-freedom spindle errors), the rod lens is shifted in the XY plane and the equality of the light intensity measured by each element of each photodiode is no longer assured. For example, if the rod lens is shifted in the +X direction, then $I_{RX1} = I_{RX2}$ and $I_{RY1} > I_{RY2}$. As a result, the direction and magnitude of spindle runout can be ascertained. The output signals, I_{RX} and I_{RY} , of the rod lens in the X and Y directions are defined by Eqs. (1) and (2), respectively:

$$I_{RX} = I_{RX1} - I_{RX2} \quad (1)$$

$$I_{RY} = I_{RY1} - I_{RY2} \quad (2)$$

Similarly, the output signals, I_{BX} , I_{BY} and I_{BZ} , of the ball lens in the X, Y, and Z directions are defined by Eqs. (3), (4), and (5), respectively:

$$I_{BX} = I_{BX1} - I_{BX2} - I_{BX3} + I_{BX4} \quad (3)$$

$$I_{BY} = I_{BY1} - I_{BY2} - I_{BY3} + I_{BY4} \quad (4)$$

$$I_{BZ} = I_{BZ1} - I_{BZ2} - I_{BZ3} + I_{BZ4} \quad (5)$$

Because of spindle errors, the output signals, I_{RX} , I_{RY} , I_{BX} , I_{BY} , and I_{BZ} , change when the rod lens and ball lens displace. The radial-motion errors, ϵ_X and ϵ_Y , are calculated using I_{BX} and I_{BY} , respectively, and the axial-motion error is

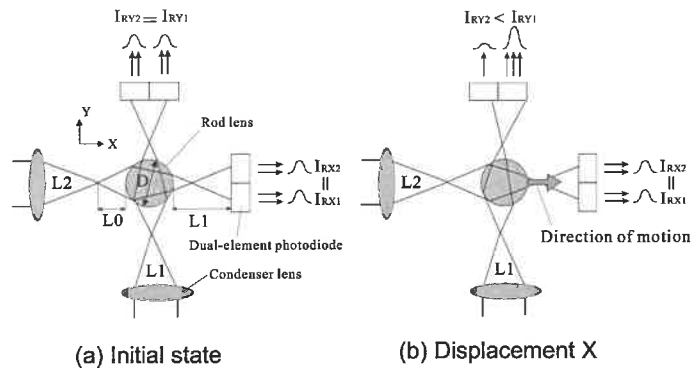


FIGURE 2. Measurement principle.

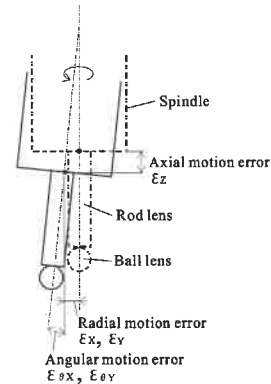


FIGURE 3. Definition of five-degrees-of-freedom spindle motion errors.

calculated using I_{BZ} . The angular-motion errors are calculated using the I_{RX} , I_{RY} , I_{BX} , I_{BY} and the distance L between the laser-irradiated part of the rod lens and the laser-irradiated part of the ball lens (Fig. 1). The distance L is 5 mm.

ERROR ANALYSIS

Due to the centrifugal force by the eccentricity between the spindle chuck and the rod (ball) lens, the rod lens is bent. As the rotational speed increases, the measurement accuracy decreases. This effect has previously been shown [3]. Also, when the rod lens and the ball lens displace, the measurement accuracy is affected by roundness of the rod lens and sphericity of the ball lens. In this study, measurement error caused by roundness of lens shape is analyzed using ray tracing. As shown in Fig. 4, the cross-sectional shape of the rod lens is considered the elliptical cross section. In case that roundness of the elliptical cross section is 0.01, 0.1, and 1 μm , the measurement error is analyzed. When the rod lens rotates from $\theta = 0^\circ$

to $\theta = 360^\circ$, the change of the output signal I_{RY} in the Y direction is examined.

For this simulation, the intensities detected by each photodiode are defined as relative intensities for all radiation rays and are expressed as I_{RX} (%), then I_{RX} (%) is converted into the displacement (μm). If there is not form error of the rod lens, the output signal I_{RX} equals 0. First, random numbers with a Gaussian distribution are generated using the Box-Muller method, and the ray based on this distribution is radiated from the condenser lens. The rays are then traced through the optical system, and the intensities expected at each photodiode are calculated. Figure 4 shows a ray trajectory of the cross section of the rod lens irradiated in the XY plane. Figure 5 shows the analytical result. The horizontal axis shows the rotation angle of the rod lens, and the vertical axis shows the displacement detection error I_{RY} caused by form error of the rod lens. The maximum errors in case of roundness = 0.1, 0.5, and $1\ \mu\text{m}$ are 0.04, 0.18, and $0.36\ \mu\text{m}$, respectively. Thus, the form error of the rod lens affects the measurement accuracy. In order to reduce the measurement error, it is necessary to use the high-precision lens or compensate the form error of the lens. The following description shows the above-mentioned displacement conversion methods from I_{RX} to the displacement. Figure 6 shows the experimentally measured change in I_{RX} and I_{RY} as a function of the rod lens displacement in the $\pm X$ direction. As shown Fig. 6, the minimum and maximum values of the output signals I_{RX} and I_{RY} are about $\pm 3\text{ V}$, and these values are equivalent to the $I_{RX} = \pm 100$ (%). Therefore, if the I_{RX} (%) is converted into a voltage value, it is about $0.03\ I_{RX}\text{ V}$. I_{RX} is linear in displacement over approximately $\pm 5\ \mu\text{m}$, and the slope of this line is $1.829\ \mu\text{m/V}$. As a result, if I_{RX} is converted into the displacement, it is about $0.055\ I_{RX}\text{ V}$.

MEASUREMENT EXPERIMENT

We performed an actual measurement to demonstrate the applicability of the system. The photodiode voltages were simultaneously recorded by an A/D card with a maximum sampling frequency of 10 MHz. For this experiment, the error motions were measured at a rotational speed of 150 krpm. Figure 7 is a photograph of the experimental apparatus used to measure the five-degrees-of-freedom error motions. Also, to make a comparison, the ball lens was measured by using the measurement

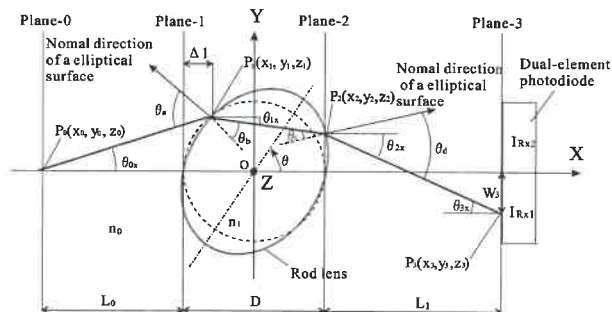


FIGURE 4. Ray trajectory of optical system.

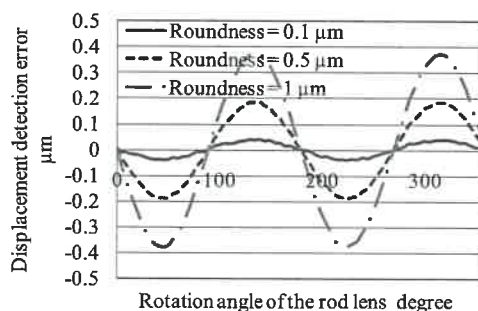


FIGURE 5. Simulation result for a roundness of the rod lens = 0.1, 0.5, and $1\ \mu\text{m}$.

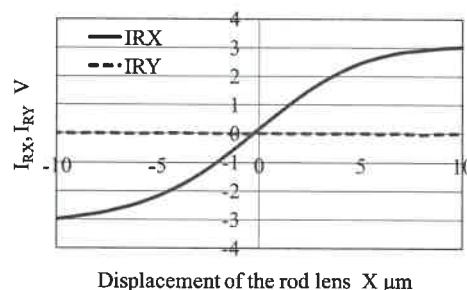


FIGURE 6. Output voltage I_{RX} , I_{RY} (+X displacement of the rod lens).

system and a commercial capacitance displacement meter. Figure 8 shows the photograph of the experimental apparatus using the capacitance displacement meter.

Figure 9 (a) and (b) show the 5 revolutions of measurement result of the radial error motion at a rotational speed of 150 krpm in the X direction by the measurement system and the capacitance displacement meter, respectively. Though it is not possible to accurately compare these measurement results since the both measurement was not performed at the same time, these shape and amplitude correspond well with each other. Because there is some install



FIGURE 7. Measurement system for the spindle error motions.

error caused by lens chucking error and adhesion error of the rod and ball lenses between the spindle chuck and the ball lens, a once-per-revolution sine curve is included in the measurement data. This sine curve is not rotation error. Therefore, this sine curve is removed by least square method. Figure 9 (c) and (d) show the measurement result after compensation. Figure 10 (a) and (b) show the 5 revolutions of the measurement result of the axial and angular error motions at a rotational speed of 150 krpm, respectively.

CONCLUSION

We have developed a measurement system to measure five-degrees-of-freedom error motions for a high-speed microspindle, and we revealed the influence of lens form error using ray tracing. In addition, by using the system for actual measurement, we demonstrate the utility of the system to measure spindle rotation errors.

ACKNOWLEDGMENTS

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FIGURE 8. Experimental apparatus using a capacitance displacement meter.

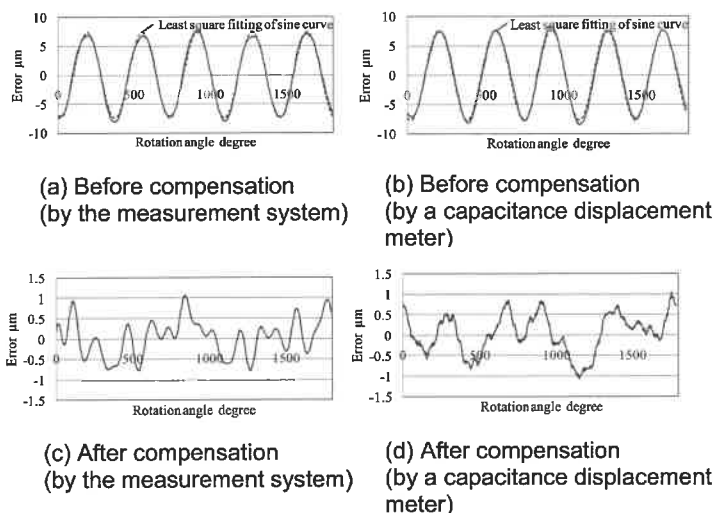


FIGURE 9. Measurement result of the radial error motions at a rotational speed of 150 krpm in the X direction.

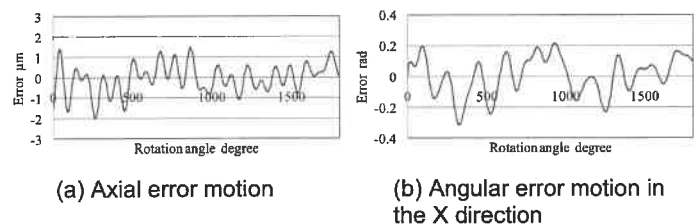


FIGURE 10. Measurement result of the axial and angular error motions at a rotational speed of 150 krpm.

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NON-CONTACT DIMENSIONAL MEASUREMENTS OF BIPOLAR FUEL CELL PLATES

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INTRODUCTION

In-process non-contact dimensional measurement is critical to the eventual rapid manufacture of fuel cell plates. This need has been identified by the Department of Energy's Hydrogen Program, and by numerous manufacturers' of fuel cell plates. To facilitate this effort, NIST has explored the use of non-contact systems for performing dimensional measurements so that manufacturers may perform 100 % part inspection during the manufacturing process.

This paper describes a dedicated system developed at NIST for performing non-contact dimensional measurements of fuel cell plates using laser spot triangulation probes. This system utilizes two probes tilted in opposing directions so that each probe can detect one side wall of the channel and some portion of the channel bottom. Procedures have been developed to calibrate the system parameters (tilt angles of the probe, offsets between probes, etc.) so that the data from the two probes can be tied to a single coordinate system. A brief description of the system and preliminary measurement results are presented in this paper.

DUAL-PROBE SYSTEM

The configuration of the system is shown in Fig. 1. The device consists of two laser spot probes and a precision translation stage with a high accuracy encoder, whose axis of motion is along the X direction. The data acquired from each probe is triggered based on stage position, and as mentioned earlier, the six system parameters (shown in Figs. 2 and 3), are used to tie the two data sets into a common frame.

The procedure to calibrate these parameters and their uncertainty are described in [1]. One approach to estimating the tilt angles (angle θ in Fig. 2) in the probes is by measuring calibrated step heights. However, this method can produce

large uncertainties in the tilt angles and does not separate the effect of the misalignments (tilt in the YZ plane, see Fig. 3) in the probe. Therefore, the tilt angles are calibrated by measuring profiles on vertically oriented gage blocks (gaging surface is parallel to the YZ plane) [1].

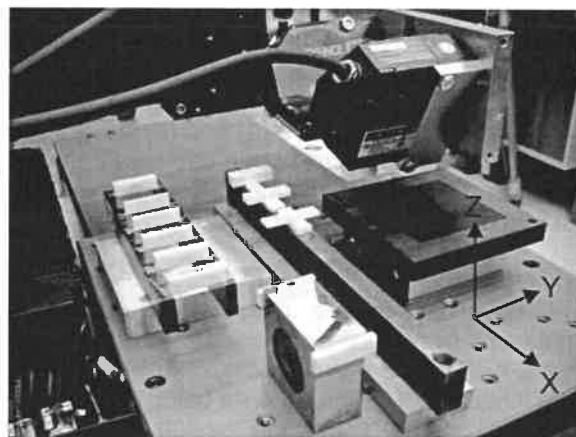


FIGURE 1. The dual-probe system.

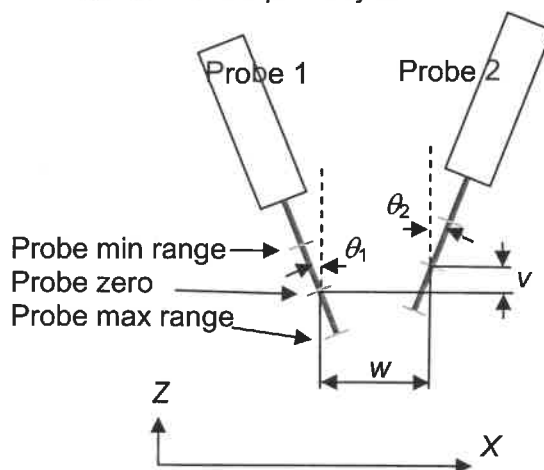


FIGURE 2. Four system parameters – tilt angles θ_1 and θ_2 , and offsets v and w . The travel axis is along the X direction.

The horizontal offset between the probes can be calibrated by measuring gage blocks of known width. But it should be pointed out that laser

triangulation probes are extremely sensitive to material properties [2] and therefore the offset calibrated using ceramic gage blocks may not be valid for other materials. When measuring fuel cell plates, a master plate of identical material as the part is used to determine this offset.

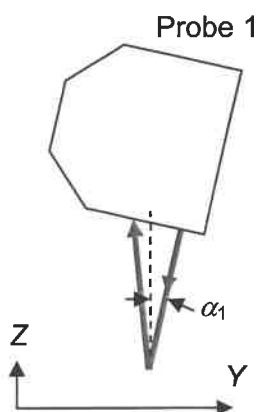


FIGURE 3. Fifth system parameter – misalignment angle α_1 . The sixth system parameter is the misalignment angle for the second probe which is not shown here.

The system was validated by measuring the height and width of numerous gage blocks. The errors in width and height measurements using the non-contact system are less than $\pm 1 \mu\text{m}$ (note that the nominal values of the gage blocks are considered the true values and the error is defined as the difference between the measured value and the true value); this suggests that channel widths and heights can in fact be measured with an uncertainty that is potentially comparable to that obtainable on a coordinate measuring machine (CMM), which is on the order of a micrometer.

PROFILES FROM FUEL CELL PLATES

Profiles from numerous fuel cell plates were acquired and the channel width and height values were compared against measurements made on a CMM with a contact probe. Some of those results are presented here.

The initial investigation was concentrated on graphite plates with vertical side walls. Fig. 4 shows a profile across a channel on such a plate. The measurements were made at 30 mm/s with a sampling interval of $3 \mu\text{m}$. For this study, pairs of plates made of identical material were considered. The plates contained 45 rows (three parallel channels that fold over

15 times), each of which was measured on a contact CMM. One plate was used as the master (to calibrate probe offset) and the other as the test artifact. The errors in width and height on the test artifact were within $\pm 2 \mu\text{m}$ of the calibrated values, indicating that the non-contact system can indeed perform high accuracy measurements, but at a much faster rate. The manufacturers of bipolar plates desire a tolerance of less than $10 \mu\text{m}$ on these features, but they have so far been unable to realize such low uncertainties with commercially available non-contact systems.

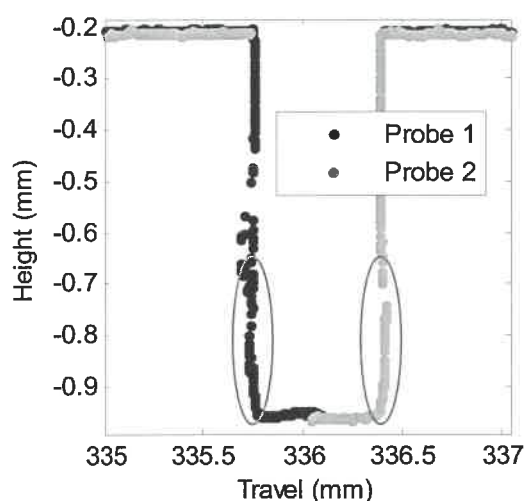


FIGURE 4. Profile across a channel of a graphite fuel cell plate with vertical walls.

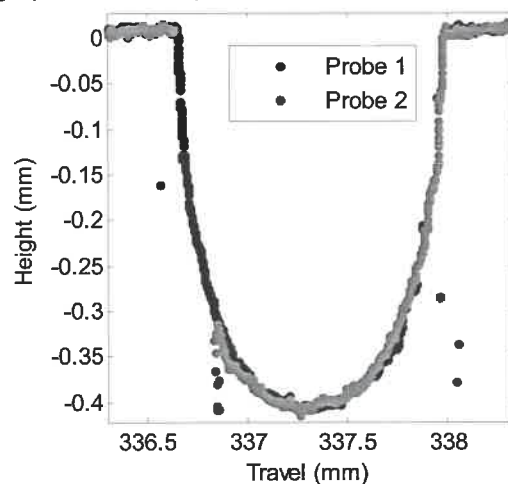


FIGURE 5. Profile across a channel of a chemically etched fuel cell plate.

Notice in Fig. 4 that the sidewall profiles near the intersection of the bottom surface are not vertical. This is really not a surface feature, but is due to multiple reflections that are possible

near the intersection of surfaces. This is a known problem with laser triangulation probes [2].

Plates made of Aluminum that were chemically etched were also measured. The profiles on one channel for two different plates are shown in Figs. 5 and 6. Notice that in one case (Fig. 6), there appears to be a dimple in the side wall, which presumably is the result of changes in the manufacturing process parameters. This is an example where rapid metrology feedback is critical in ensuring process stability.

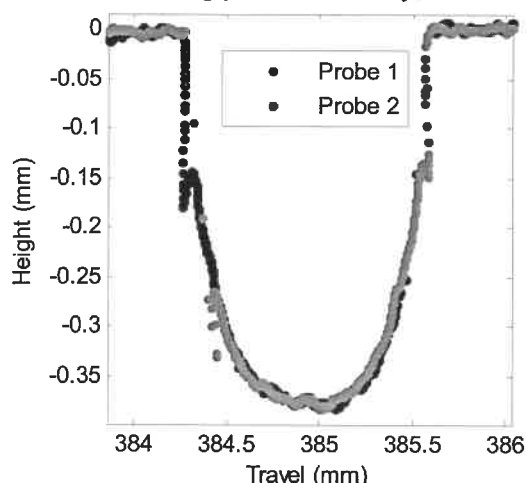


FIGURE 6. Profile across a channel of a chemically etched fuel cell plate.

Detailed uncertainty budgets for channel height and width for the case of the graphite fuel cell plates with vertical channel sidewalls have been developed. The calculations, described in [1], show that the expanded uncertainty in width is $6\text{ }\mu\text{m}$ ($k = 2$) while that of height is $3.8\text{ }\mu\text{m}$ ($k = 2$).

SPEED TESTS

In-process measurement that supports rapid manufacturing requires that measurements be made not only in-situ, but also in a timely manner. The system described here relies on acquiring profiles on plates by scanning the probes across the part. The initial experiments were performed at 30 mm/s, where a single profile across a 100 mm long plate would take a little more than 3 s. Acquiring six parallel profiles spread out on the plate would therefore take about 20 s.

Measurements of a single fuel cell plate at different speeds, from 30 mm/s up to 500 mm/s, were performed to evaluate if there is any noticeable degradation in accuracy at higher

speeds. The channel heights and widths at different speeds for each of the different rows are shown in Figs. 6 and 7 respectively. Notice that the channel heights and widths at 300 mm/s agree with those at 30 mm/s rather closely. Some discrepancy is observable at higher speeds, but for the most part, the agreements in width and height between 500 mm/s and 30 mm/s are quite consistent.

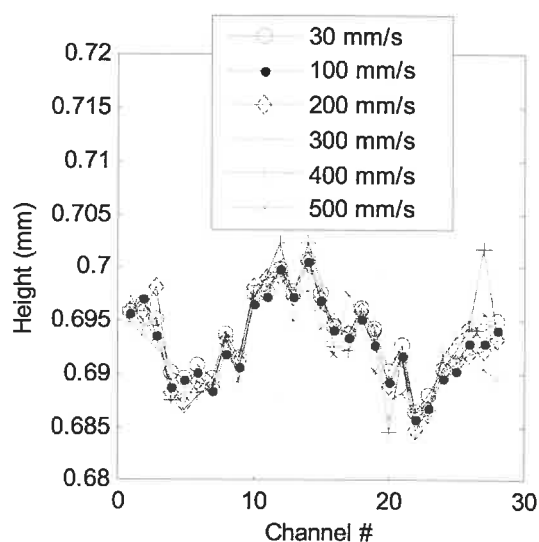


FIGURE 6. Channel heights as a function of channel number at different speeds.

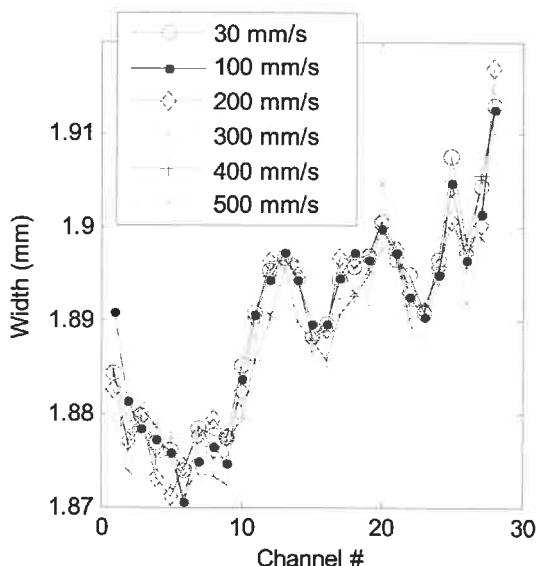


FIGURE 7. Channel widths as a function of channel number at different speeds.

Fig. 8 shows a profile across a few channels acquired at a speed of 30 mm/s. Fig. 9 shows a profile across the same channels at 500 mm/s. The lower data density at 500 mm/s (sampling

interval of 25 μm as opposed to 3 μm at 30 mm/s) was necessitated by the hardware used. In addition, it can be seen that there are numerous outliers in the profile which have to be carefully processed prior to any numerical computations, such as height and width of channels. However, as Figs. 6 and 7 show, careful processing of the data acquired at 500 mm/s can provide excellent agreement with that acquired at 30 mm/s.

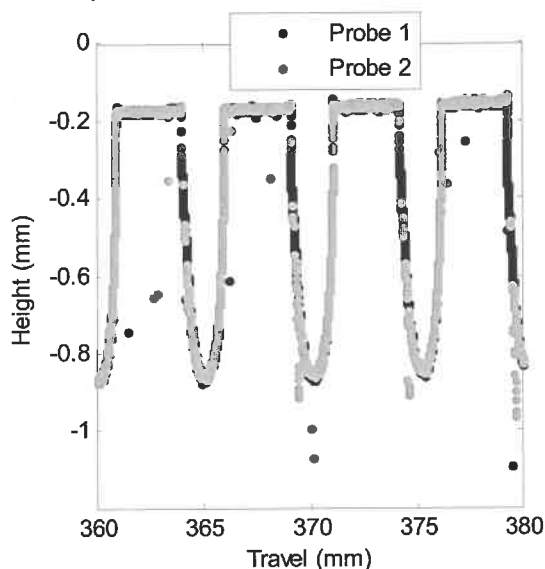


FIGURE 8. Profile across a few channels at 30 mm/s.

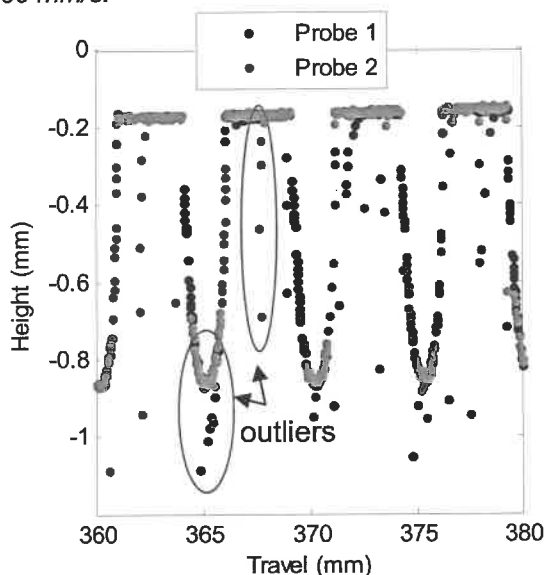


FIGURE 9. Profiles across a few channels at 500 mm/s.

CONCLUSIONS AND FUTURE WORK

This paper describes a dedicated non-contact fuel cell measurement system. A unique aspect

of this design is the use of two tilted probes so that information from the side walls of channels can be captured, in addition to the top surface of the plate and the bottom of the channel.

Preliminary measurement results obtained from the non-contact system are described in this paper along with a brief summary of some of the challenges in using laser triangulation probes. One of the primary barriers to the adoption of such non-contact systems towards in-process measurement is the measurement speed. Experimental evidence has been presented here that demonstrates the non-contact system can measure channel height and width even at speeds of 500 mm/s with little degradation in accuracy.

Some anticipated future work is to modify the setup towards measurement of plate thickness by placing one probe above the plate and the other below. Thickness and parallelism measurements are critical towards the concept of "smart assembly" [3], which is to tag each plate with its own thickness and parallelism information so that during assembly, appropriate plates can be chosen to improve the overall stack's parallelism.

ACKNOWLEDGEMENTS

The authors gratefully acknowledge the assistance of Patrick Egan at NIST in this work, and also thank Keith Stouffer and Vincent Lee for reviewing this paper.

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Metrology of a Precision Plasmonic Lithography Stage

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Abstract

Plasmonic lithography is a promising technology for next generation nanoscale manufacturing providing a solution of semiconductor industry to continue to shrink linewidths to less than 20 nm. To achieve these dimensions it is necessary to integrate all the components into a whole manufacturing system. This system needs to be validated metrologically. In this paper, a thorough metrology scheme is proposed to model, measure, calibrate and correct the systematic errors of the stage's positioning performance. Capacitance gages and interferometers provide the critical assessments are used for geometric errors. Thermal and environmental deviations will be studied separately.

A detailed systematic error model based on homogeneous transformation matrices (HTMs) has been derived.

Keywords:

Nano-positioning, Precision metrology, Error budget, Calibration, Plasmonic Lithography

Introduction

A novel plasmonic imaging lithography machine is being developed under the Center for Scalable and Integrated NanoManufacturing (SINAM [1]) to meet 10nm linewidth critical dimension. A solid model of this is shown in Figure 1.

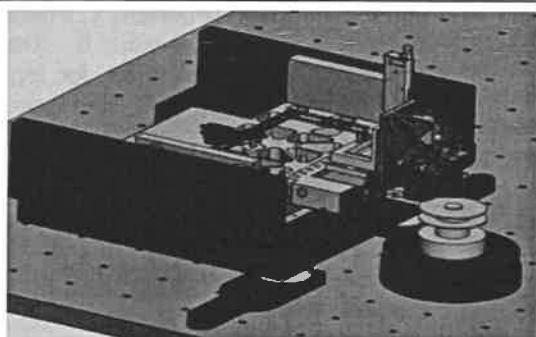


Figure 1. Overview of PIL machine (courtesy of Mohamed Saad)

The main stage of the machine is driven by two linear motors whose movement is measured by two optical scales on either side. A 3D pre-focusing stage is designed to guide the laser pulse to focus on the plasmonic imaging lens having a vertical clearance of about 10 nm relative to the disk. In Figure 2, the stage has motion in one direction towards the center of

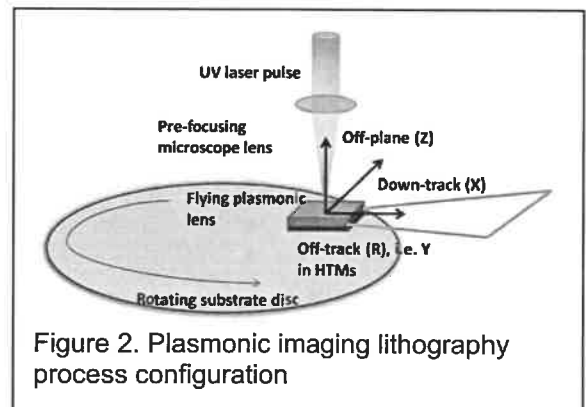


Figure 2. Plasmonic imaging lithography process configuration

disk across a range of 35 mm.

A metrology system was built for repeatable error compensation. It has been modeled using HTMs. Separate experimentation methods are used to map geometric errors: linear positioning, out of straightness, pitch and roll.

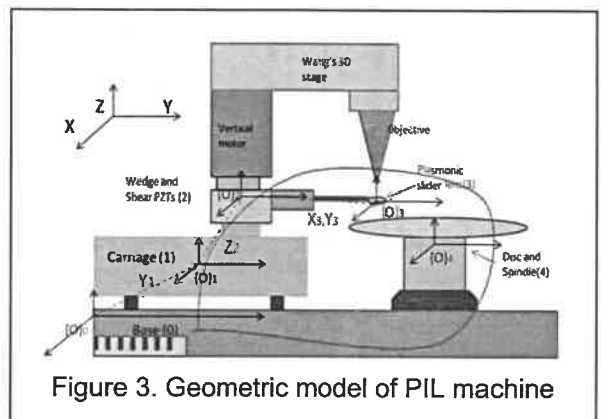


Figure 3. Geometric model of PIL machine

HTM modeling

Based on a rigid body model of the machine structure, the geometric relationship is vector modeled as shown in Figure 4 which does not include the errors of the spindle.

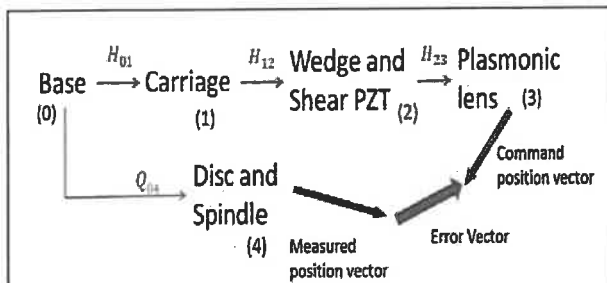


Figure 4. Geometric relation between HTMs

A metrology model can be abstracted from the metrology loop and HTM models. The coordinate transformation relation can be derived from the subscripts of the HTMs. The

$$Q_{04} = \begin{bmatrix} 1 & 0 & 0 & d + thermal_d \\ 0 & 1 & 0 & r + thermal_r \\ 0 & 0 & 1 & h + thermal_h \\ 0 & 0 & 0 & 1 \end{bmatrix}$$

Figure 5. Error propagation from base to spindle system

modeled error of concern is the deviation of real position from the command position as indicated by the red arrow in Figure 4.

The corresponding mathematical relation from the base to the lens and spindle is illustrated in Figure 5 through 7 in matrix form.

$$E_{(0)} = P_{(0)} - Q_{(0)} = \begin{bmatrix} \delta_x(Y_1) + \varepsilon_x(Y_1) \cdot Z_2 - thermal_d \\ \delta_y(Y_1) + \delta_z(Z_2) - \varepsilon_y(Y_1) \cdot Z_2 + \alpha_{y,z_2} \cdot Z_2 + thermal_x - thermal_y \\ \delta_z(Z_2) + \delta_x(Y_1) - thermal_h \\ 0 \end{bmatrix}$$

Figure 6. Terms of error correction

$$H_{04} = \begin{bmatrix} 1 & -\theta_z - \varepsilon_z(Y_1) & \varepsilon_y(Y_1) & \delta_x(Y_1) + \varepsilon_x(Y_1) \cdot Z_2 \\ \theta_z + \varepsilon_z(Y_1) & 1 & -\varepsilon_x(Y_1) & Y_1 + \delta_y(Y_1) + \delta_z(Z_2) - \varepsilon_y(Y_1) \cdot Z_2 + \alpha_{y,z_2} \cdot Z_2 + thermal_x \\ -\varepsilon_y(Y_1) & \varepsilon_x(Y_1) & 1 & Z_2 + \delta_z(Z_2) + \delta_x(Y_1) \\ 0 & 0 & 0 & 1 \end{bmatrix}$$

Figure 7. Error propagation from base to lens

Measurement List and Plan

To obtain all the listed error terms, specific measurement plans are proposed in Figure 8. The experimental configurations are given as follows from Figure 9 to 12. A plane mirror interferometer [2] is built to meet high stability

Type	Symbol	Measurement plan
Linear positioning accuracy	$\delta_y(Y_1)$	Plane mirror Interferometer (direct)
X straightness of Y of carriage	$\delta_x(Y_1)$	Single cap gage (direct)
Z straightness of Y of carriage	$\delta_z(Y_1)$	Single cap gage (direct)
Roll angle of Y axis of carriage	$\varepsilon_y(Y_1)$	Double cap gages (differential manner)
Pitch angle of Y axis of carriage	$\varepsilon_x(Y_1)$	Triple cap gages (combined differential manner)
Squareness of Z and Y of wedge	α_{y,z_2}	Observed and estimated by autocollimator (indirect)
Straightness of Z axis of wedge	$\delta_z(Z_2)$	Integrated with the Z straightness of Y in carriage
Straightness of Y axis of wedge	$\delta_y(Z_2)$	Integrated with linear positioning accuracy

Figure 8. List of error terms and measuring methods

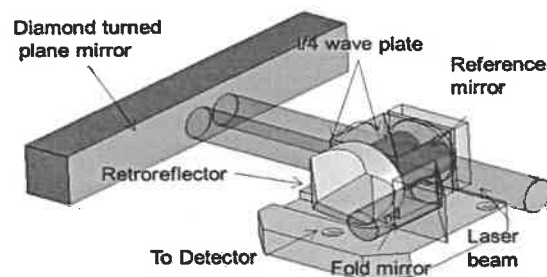


Figure 9. Plane mirror interferometer (courtesy of O. Ozturk)

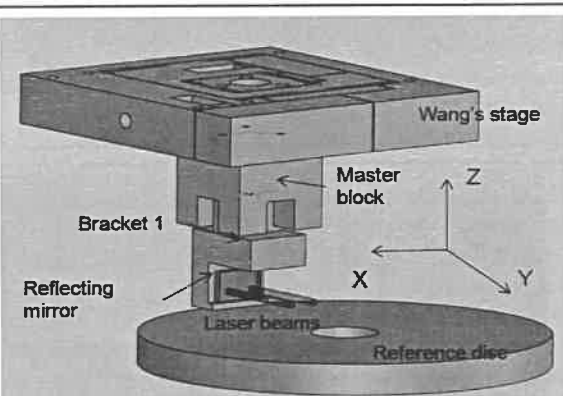


Figure 10. Linear positioning measurement

requirement. Capacitance gages are used to measure small range displacements which can be transformed to out of straightness, roll and pitch angle. In particular, straightedge reversal technique is used to study the off track deviation error induced by stage's moving carriage [4].

The independent measurement units illustrated before are attached to the master block which is bolted to the 2D flexure stage designed by Dr. Wang [5].

Before the stage is built, a test system is set to validate the measurement plan on a Moore Measuring Machine as shown in Figure 13.

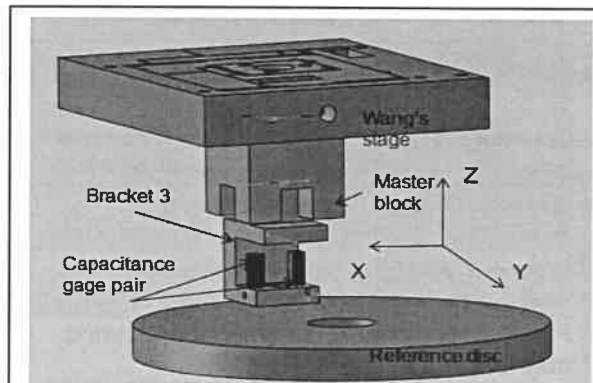


Figure 11. Pitch, roll and Z straightness measurement. Bracket 3 is rotated 90 degrees for pitch measurement.

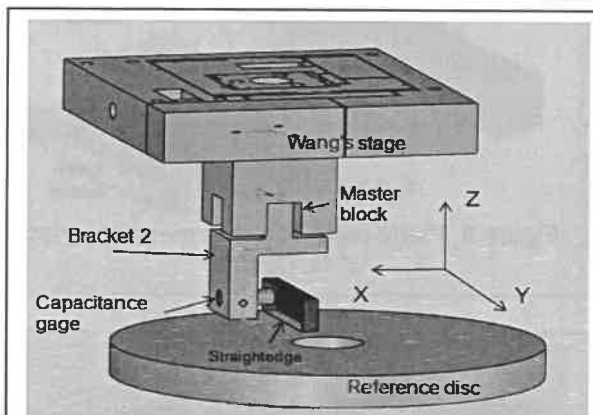


Figure 12. X straightness measurement

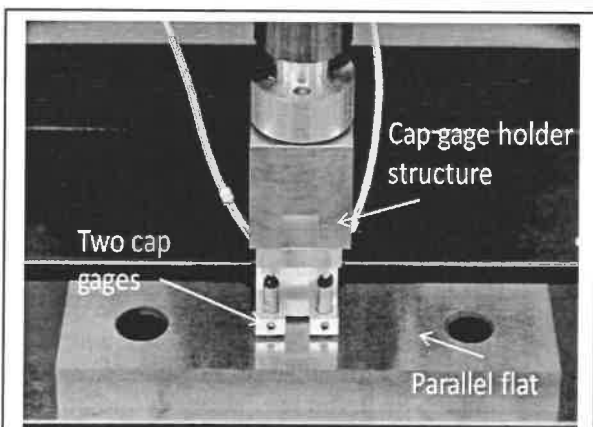


Figure 13. Test system setup

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MEASURING POSITIONING ACCURACY OF LARGE MACHINE TOOLS

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SUMMARY

Nowadays, manufacturing of large parts is a driving force in machine tool industry. This demand has challenged machine tool builders to materialize designs in a new scale and even more, to keep the accuracy of these machines at the level of smaller machines.

On the other hand, builders lack appropriate tools to assess positioning accuracy of big machines in practical ways. The classical set of instruments used to setup the machine, such as precision levels, squares or linear interferometers with different kinds of optics is becoming obsolete when addressing this new family of machines.

The way to overcome these difficulties seems to be grounded on the way instruments are used more than in the development of new instruments. The base measuring technology best suited for this task seems to be laser interferometry. Other devices such as Ball Bars are able to provide accurate measurements although they get impractical when verifying very large machines.

Nowadays more and more workshops have access to Laser Tracker devices. These are very handy during machine tool assembly or during the setup of jigs; on the other hand, they lack accuracy to verify machine tool positioning accuracy. Lately several approaches have arisen that take advantage of the high accuracy of the laser interferometer while disregarding the information from the rotational encoders. The basic idea relies on combining linear interferometer measurements from sequential measurements in a multilateration scheme [3]. However, when experimenting with very large machine tools some limitations become evident. In house experiments have shown that the main obstacle to achieve the limiting accuracy of this approach is instability of the ambient conditions and machine tool thermal behavior. Due to these factors, accuracies expected from pure multilateration become meaningless.

In this work an intermediate approach has been developed, where linear and rotational measurements are merged, with the aim of reducing the time required by the

measurements. The benefit is double, being the measurement more economical and the machine suffering smaller thermal drifts during measurement.

So the technique is valid to provide a positioning error map accurate enough to enhance the performance of the machine in future mechanical or numerical compensations and even to provide a feedback to the design.

INTRODUCTION

Usually the measurement of positioning errors of machine tools has been made in different stages, separating errors of linear positioning, straightness and orthogonality. This procedure was followed due to the available measuring devices.

Since the introduction of multilateration techniques, 3D positions in space have been able to be measured with enough accuracy, so the former procedure is no longer required. In addition, usually multilateration techniques are practical enough to provide information on the complete working volume of the machine, not only on selected positions, lines or planes.

There are different procedures to measure translational and rotational movements [1],[2].

On the one hand, complete 6 dof information can be gathered measuring distances from fixed points, the instrument positions, to different points that move jointly with the tool. It can be shown that at least, height sets of measurements are required from four instrument positions to three points that move jointly with the tool.

On the other hand, an almost complete set of measurements can be obtained using multilateration to measure the translations of the tool and electronic levels to measure rotational movements along two horizontal axes. In this article, some interesting advantages of this procedure are shown.

MULTILATERATION AND LEVELS

In the following, the advantages of using a multilateration technique to measure positions and electronic levels to measure two angular deviations are depicted.

In practice, multilateration measurements are carried in a sequential scheme. This is because only one measuring device is available. So machine movements are repeated several times and measurement are taken from different positions around the working volume, instead of measuring simultaneously all of them. In order to measure only positions, four measurements are required. In figure 1 the machine, the working volume and the measuring device are shown.

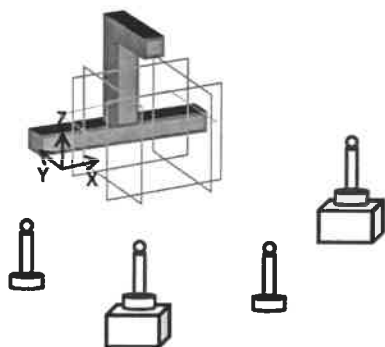


FIGURE 1. 3 dof multilateration

Therefore, for measuring the spatial deviation of a big machine-tool, it is required a long time elapse (several hours) and machine stability (e.g. thermal deviation) can limit substantially the measurement accuracy.

On the other hand, the use of electronic levels provides direct measurement of the magnitude of interest, that is, the angular deviations of the tool. Nowadays wireless electronic levels do exist, so measurements along long strokes are no longer a problem. The resolution, 1 micron/m, and accuracy of state-of-the-art commercial levels are good enough to verify large machine tools.

The level measurements can be recorded in every single repetition of the machine movements. So they not only provide the measurement of interest, but they also are a very good indicator of the stability of the machine during the measuring time. This last point is very important, because the multilateration measurements are reliable only if the machine has not suffered from drifts.

Another great advantage of using the levels lies in reducing measurement time, because only four repetitions are needed instead of the eight.

Time is a crucial factor that affects two different aspects of the measurements, listed as follows:

i) Machine schedule time: it is convenient taking the machine out of working schedule to a

minimum, as every hour the machine is not working is money spent.

ii) Machine stability: as environmental factors may change during measurements (e.g. room temperature), making short time measurements ensures minimal thermal deviation of the machine that could introduce inaccuracies in results. So using the levels, not only provides an indicator of machine drifts as said previously, but it also helps to keep them small.

In figure 2 a time schedule comparison between different methods of measuring volumetric deviations is shown. Some temperature profiles are also shown, although they correspond to specific measurements.

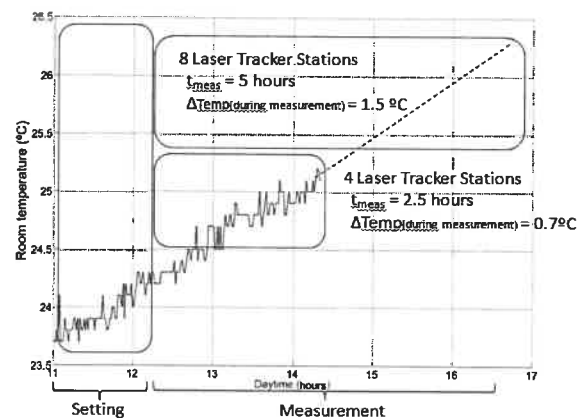


FIGURE 2. Measurement time schedule

EXPERIMENTAL RESULTS

The instrumentation used to perform the measurements consists on:

i) Leica Absolute Tracker AT901MR: The Laser tracker is a very powerful instrument which combines laser interferometer and encoder measurements to provide 3D translational measurements of the provided reflector. However a quite important weakness is that measurements are not accurate enough to verify machine tool positioning. The limitation lies on the use of rotational encoders, on the other hand, the interferometric measurements are very precise.

ii) WYLER BlueLevel: These are state-of-the-art precision electronic levels. Beside being very accurate, they provide a very interesting feature, they send the measurement by radiofrequency to a remote meter, so long range movements are not a problem. On the other hand, stabilization time of the measurement is in the 5 seconds range, so no time is lost due to this stabilization issues.

In figure 3 a ZAYER milling machine model 30KCU2600R is shown, which was used to perform measurements using the methodology described.



FIGURE 3. ZAYER 30KCU2600R milling machine

In figure 4 two photographs of the measurement setting are shown. It can be seen the measured milling machine, the Laser Tracker used, and the reflector and precision wireless levels put on the headstock.

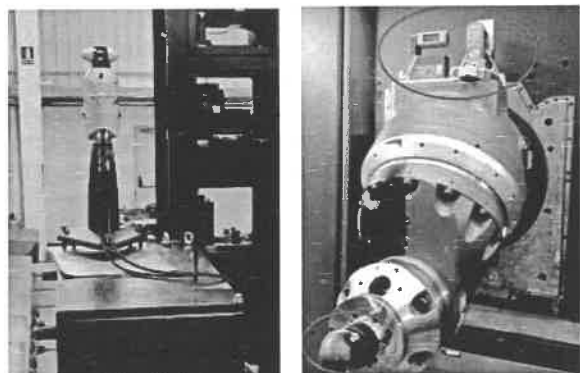


FIGURE 4. Measuring equipment

Figure 5 shows the measurements of one of the levels along the four repetitions. The good repeatability shows that no critical machine drifts have appeared during the 2.5 hours of measurement.

Measured volume was
 along longitudinal guide → 2000mm
 along headstock stroke → 2000mm
 along column height → 5000mm

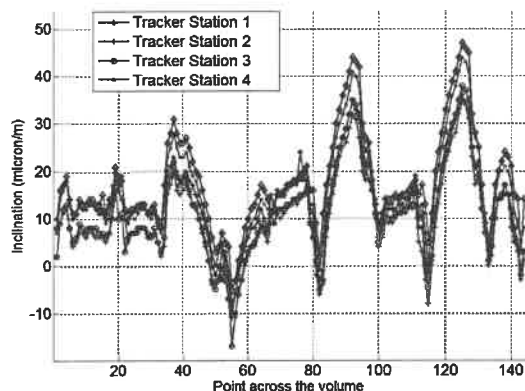


FIGURE 5. Level measurements

No longer stroke along the longitudinal guide has been considered because the interest was focused in the YZ plane behavior.

Altogether the positioning in 144 points was measured. These points were distributed in four different vertical planes. Figure 6 shows the result obtained by multilateration. In this figure, the errors have been amplified. The scale of the different axes has not kept real, as stroke in vertical direction is 5000 mm. Two representations are shown, a general 3D view and a comparison of the two YZ planes.

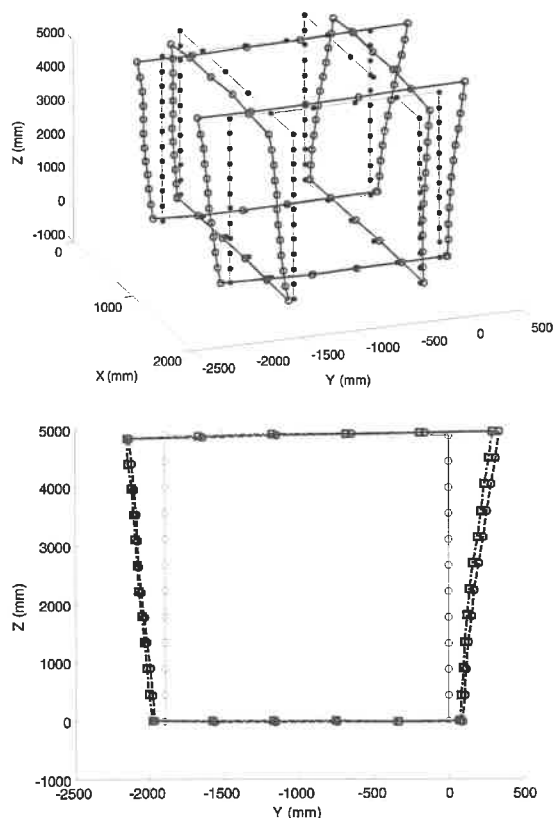


FIGURE 6. Volumetric measurements

There are several factors that prevent from fitting exactly the systems of equations that provide the 3D positioning of the machine:

i) The multilateration measurements provide some kind of redundancy when more measuring points than the minimum are used. A minimum of 4 measuring stations a minimum of 6 measuring points are needed.

ii) Measurements are not perfect. Interferometer measurements are under the influence of ambient conditions. As these change or are somehow unknown the measurements will be inaccurate.

iii) The machine does not provide absolute repeatability. The most limiting factor is the thermal behavior of the machine under the influence of ambient temperature.

The misfit of the system of equations provides a good indication of how well the measurements have been made. In figure 7, a histogram of the misfit of the measured lengths is shown. In order to provide more redundancy to the measurements, and gain confidence in the results, a 5th measuring station has been used. The figure shows the misfit histogram for both four and five multilateration measurements. It can be seen that almost all the measurements show less than 5 microns discrepancy. This reveals that machine repeatability is good and that measuring conditions are appropriate.

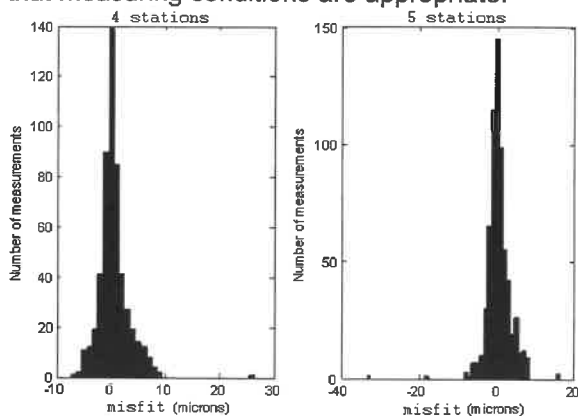


FIGURE 7. Misfit histograms

Figure 8 shows a comparison of rectitude measurements of two different movements along the 5000 mm Z stroke. It can be seen that the machine behavior is similar on both movements, and that the measurements are accurate enough to study 10 micron rectitude errors along 5000 mm distances.

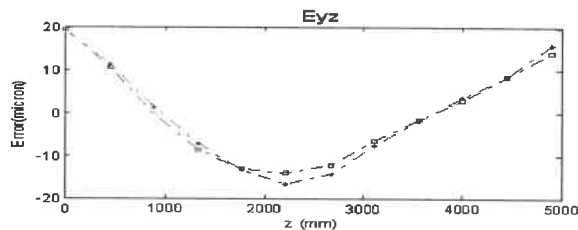


FIGURE 8. Rectitude comparison

FUTURE WORK

The next works will be focused in the development of measurement procedures to be able to measure the third rotational dof via multilateration techniques while the other two will be measured by electronic levels.

CONCLUSIONS

In this report the advantages of merging multilateration results and electronic level measurement to verify large machine tool positioning has been described. They can be summarized as:

i) Measuring rotational dof with electronic levels allows for dramatic reduction in measurement time, compared to measuring them via multilateration techniques, with two benefits:

1. Machine stability: less drift.
2. Machine schedule time: less money.

ii) Electronic levels provide a good indication of machine stability during the measurements.

On the other hand, one day measurements provide valuable information to carry out mechanical or numerical compensation during machine setup or retrofitting. Measuring errors lower than 5 microns are expected when verifying 5000 mm stroke vertical columns.

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PRECISION ENGINEERING IN THE CONSTRUCTION AND ALIGNMENT OF THE NIST CALCULABLE CAPACITOR

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ABSTRACT

This paper summarizes the current NIST effort to manufacture a new calculable capacitor, focusing on issues associated with the fabrication of the four main electrodes and their subsequent assembly into a stable geometric form useful for the realization of a calculable capacitance. Results presented demonstrate how precision manufacturing, translation, and engineering design concepts have been used to create a compact, stable capacitor the alignment of which can be achieved through the use of simple optical levers.

INTRODUCTION

The NIST calculable capacitor is a unique high accuracy instrument that directly links the capacitance unit to the mechanical unit of length and serves as the fundamental realization of the Farad in the USA. It is a physical realization of Thompson and Lampard's theorem [1], and the classic design [2-4] employs four fixed cylindrical electrodes and upper and lower blocking electrodes, as shown in Fig. 1. The average cross capacitance between opposite electrodes is precisely related to the distance between upper and lower blocking electrodes, so that the problem of measuring electrical capacitance is reduced to one of measuring mechanical displacement along a single axis, which we achieve using *pm-level* interferometry [5].

A challenge in making a calculable capacitor is the fabrication of electrode bars to near-perfect cylindricity, specifically to within 100 nm over the working length. We have had some success diamond turning aluminum electrodes [6], and here we quickly review recent results using copper. We then look at precision engineering solutions used to achieve the geometry of Figure 1, framing the problem within the context of the Thompson and Lampard theorem and its inherent use of symmetry. Finally, we consider the problem of translating the blocking electrode, suggesting a solution based on commercially available vacuum compatible air bearings.

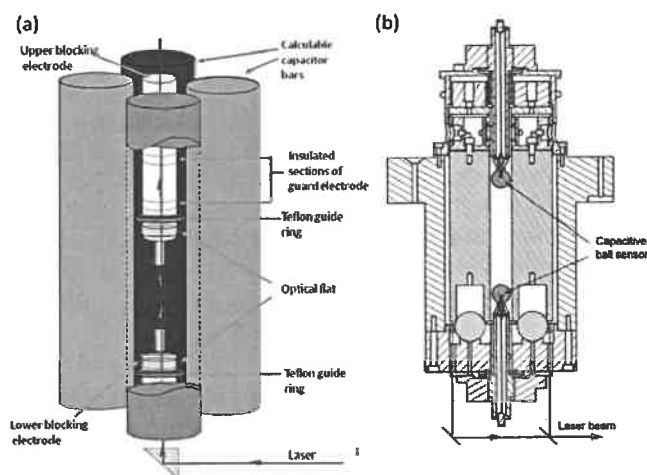


Fig 1. Calculable capacitor (a) Cartoon of essential design elements where capacitance varies as upper blocking electrode is moved; distance between upper and lower blocking electrodes is measured with Fabry-Perot interferometry (b) Cross-section of NIST prototype. Capacitor bars (blue) pivot on silicon nitride spheres (yellow). Bars can be aligned using capacitance ball sensors (red) and laser light reflected off of mirrors machined into their end faces.

ALUMINUM VS COPPER FABRICATION

Initial results reported in [6] showed that an aluminum cylinder nominally 40 mm diameter and 160 mm long could be produced without taper and with profiles at various places around its circumference that deviated from straight by less than 25 nm. Interestingly, recent attempts to extend this fabrication to a copper blank met with less success. The data obtained using the NIST M48 coordinate measurement machine revealed that the copper part tapers by as much as four micrometers along its length, so that the part is a mild cone, rather than a "perfect" cylinder, and it is substantially outside the 100 nm target. Subsequent efforts to machine a variety of different cylinders revealed that the diamond turning machine used to manufacture

the parts had drifted out of alignment. We now employ on machine metrology to characterize the machine and part errors using reversal techniques [7]. Although time consuming, this approach has once again yielded parts of near perfect geometry.

ASSEMBLING PARALLEL BARS

Four aluminum test bars have been made and assembled into a prototype calculable capacitor. Given electrodes that are cylinders (not cones), the problem of assembling and aligning a classically designed calculable capacitor reduces to ensuring that all four bars are parallel and centered about the blocking electrode's translation axis. Interestingly, it has been shown [2] that if the assembly is very nearly symmetric, then the Thompson-Lampard theory can be expanded in a series approximation revealing that deviations from perfect symmetry affect the computed capacitance in a fashion that is second order in terms of the difference in the diagonal cross capacitances. One important consequence is that the capacitance per unit length is independent of the size, and as long as thermal expansion is symmetric (the bars don't tilt too much) the capacitance per unit length remains constant.

The new NIST capacitor in Figure 1 (b) starts with a circular base plate that has four cones machined at 90° intervals at a fixed radius from the plate's center. A precision silicon nitride sphere is seated in each cone to serve as the ball in a ball and socket pivot. The socket is machined into the base of each bar, so that the final assembly is similar to having four ball bars mounted symmetrically about the center of the plate and housed in a canister that is concentric. Fine screw adjusters are included at the top of the canister to tip the bars into alignment.

A novel feature of our approach is the inclusion of alignment mirrors machined into the capacitor electrodes as part of their fabrication. The socket ends of the bars have been faced with the diamond tool to leave a reference mirror whose orientation with respect to the cylinder axis can be measured on the M48. Using this mirror surface and a laser beam aligned to a suitable reference, such as the earth's gravity, the cylinders can be adequately aligned (< 2 microradian) using simple optical levers.

The essential elements of the optical lever are illustrated in Figure 2, and the procedure for

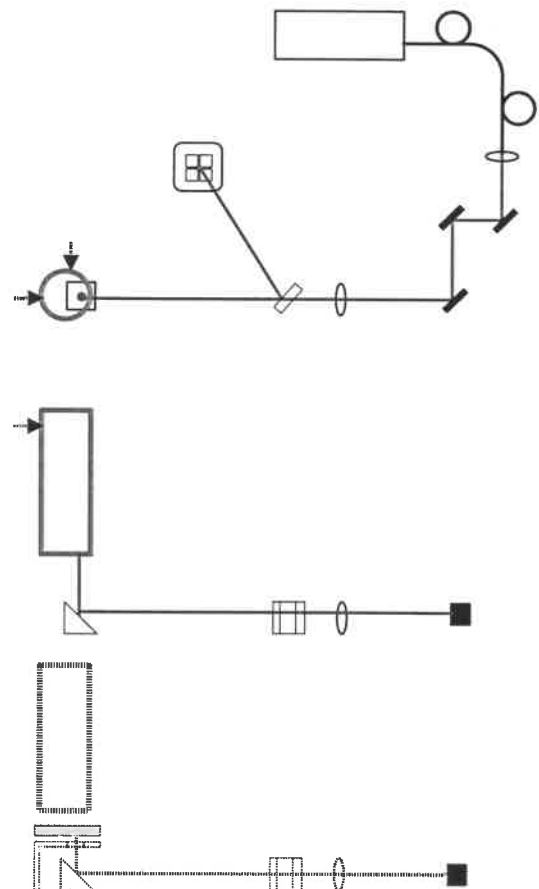


Fig. 2. Schematic of optical lever components.

alignment is as follows. A laser beam is first aligned to a reference flat that has been aligned to gravity using a suitable leveling device, such as a precision spirit level. The absolute alignment with respect to gravity is unimportant. What is important is that the aligned mirror be a stable and repeatable reference, consistent within a microradian for the alignment of each bar. In preliminary tests, we have used an optical flat with a mirror coating on one side. The mirror is placed facing downward on three sapphire balls secured to the top of a tip and tilt platform having an aperture for the laser beam, as illustrated. The flat is leveled in each axis using a bubble level having divisions corresponding to 40 arc seconds. The laser beam is then adjusted such that it returns on itself to within less than 1 mrad as measured using a small aperture at the beam launch position. Next, the portion of the returning laser beam diverted by the beam splitter is used to illuminate a quadrant photo detector. Note, that a long focal length lens is employed to bring the

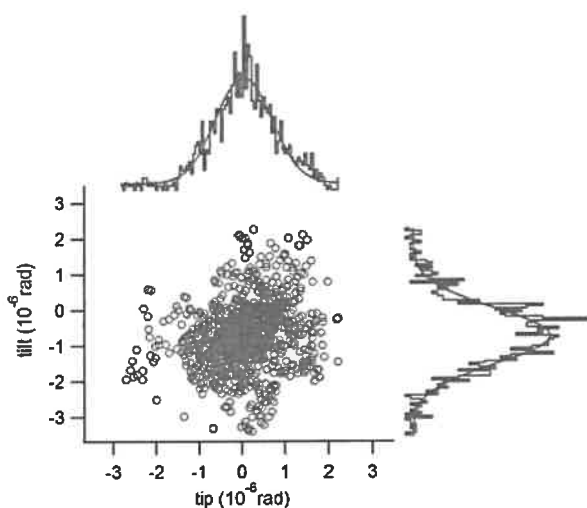


Fig. 3. Typical position noise from optical lever when aligned to main bar.

beam to a focused spot on the detector, which is positioned so that the output of the detector is null when the spot is centered. The deflection sensitivity of the photodetector is calibrated using micrometer screws to translate the detector about the null position with respect to the fixed beam. The angular sensitivity is then approximated assuming an optical lever arm of nominally one meter length. The optical flat is then removed, and the beam is steered (translations) to reflect off the mirror surface of a main bar electrode, which is then adjusted using the tip and tilt screws until the detector is again reading null. The optical flat is restored to verify that the beam steering translations did not affect the beam verticality, and final adjustments made to the beam alignment before removing the flat and performing the fine alignment of the main bar.

In experimenting with this apparatus it appears that the mirror can be lifted from the tip tilt platform and resealed without disturbing the alignment by more than a few microradians. Greater care in the development of a kinematic mount for this mirror should only improve the situation. We also observe that once aligned, the optical lever system comprising the laser, optics, detector, and main bar electrode is stable to better than a micro radian, as shown in Figure 3. The bars have been perturbed from alignment and re-adjusted back to position to within this stability limit using only the manual screw adjusters. From these experiments we conclude that the main bar support structure, the ball and

socket pivot, and the screw adjusters provide an assembly having suitable stability and range of adjustment to facilitate construction of a reliable calculable capacitor.

TRANSLATING THE BLOCKING ELECTRODE

As stated previously, all electrodes must be parallel to and centered about the translation axis of the blocking electrode. As a result, it is important that a smooth, single axis of translation be defined using a mechanism with an absolute minimum of error motion. The range of travel required is on the order of 60 mm, and, because the blocking electrode must be cantilevered off of the translation axis, angular deviations must be minimized (again, < 2 microradian). The range of motion seems too large for a flexure stage; the angular deviation specification too stringent for a roller bearing; and attacking the problem with an actively compensated, six-degree of freedom translation stage seems an unnecessary level of complexity. Instead, we are investigating whether or not the motion can be adequately constrained passively, using a somewhat unorthodox precision engineering solution: a vacuum compatible air bearing. Specifications for these devices suggest that they offer the desired range of motion and the desired level of off-axis constraint in a simple, robust, and very cost-effective package.

CONCLUSIONS

Great progress is being made towards realizing a new calculable capacitor. Our updated version of this classic electrical standard has benefitted significantly from the application of several techniques borrowed from precision manufacturing and engineering. Demonstration of the translation of the blocking electrodes and incorporation of picometer interferometry are next up, and they will present their own set of challenges to be met through careful application of precision techniques.

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SCANNING WHITE LIGHT INTERFEROMETRY FOR USE IN CHARACTERIZING BIOMIMETIC IMPLANT COATINGS

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INTRODUCTION AND BACKGROUND

Phosphatidylserine, specifically, 1,2-dioleoyl-sn-glycero-3-phospho-L-serine (or DOPS), has established itself as a promising candidate for use in biomimetic medical implant coatings for enhancing osseointegration[1]. Little is known about these soft and relatively thick surface coatings as they are difficult to characterize.

Typical biological coating characterization methods include thin film analysis techniques, scanning electron microscopy (SEM), atomic force microscopy (AFM), contact angle measurements, *in vitro* cell studies and *in vivo* animal studies[2]. Most thin film analysis techniques do not work with DOPS coatings on metallic implants due to their relatively thick and rough nature (coatings are on the order of 5-20 μm in thickness). SEM and AFM are useful but often resource intensive, require modification of atmospheric conditions, and limited in the size scale of useful information that can be measured (eg. quantifiable in only nano or only micro size scale ranges)[3]. *In vitro* and *in vivo* methods are inherently difficult, expensive, and often have gross variance in the results. Aqueous contact angle measurements are a useful way to measure surface hydrophilicity. However, effects of surface composition on contact angle are often confounded by concomitant changes in surface roughness. Thus it is desirable to be able to non-destructively measure roughness of samples prior to performing contact angle tests.

Scanning white light interferometry (SWLI) was used in the presented work to measure the surface roughness of DOPS coatings on titanium alloy ($\text{Ti}_6\text{Al}_4\text{V}$) with varying concentrations of cross linker molecules. The roughness measurements from SWLI were then compared and correlated to contact angle measurements of the DOPS coatings.

MATERIALS AND METHODS

DOPS coatings were applied to square $\text{Ti}_6\text{Al}_4\text{V}$ samples via an e-spray deposition method[4]. The e-spray process utilizes a voltage differential to atomize a liquid and efficiently coat conductive substrates[5]. Briefly, a 1.3% (vol%) DOPS in chloroform solution was e-sprayed onto cleaned and passivated 7.5x7.5mm $\text{Ti}_6\text{Al}_4\text{V}$ samples. The voltage differential used was 12 kV with a pump rate of 14 mL/min and an 8 cm working distance.

The DOPS coated samples were prepared with varying concentrations of cholesterol incorporated into the coating. Cholesterol is a known biomolecule that helps control cell membrane rigidity *in vivo*[6]. Molecular ratios of DOPS to cholesterol used for the coating groups were as follows; 100% DOPS, 24:1 (DOPS:cholesterol), 12:1, 6:1, and 1:1.



FIGURE 1. SEM of a pure DOPS coating (no Ca) on a $\text{Ti}_6\text{Al}_4\text{V}$ metal substrate. 1,000X Magnification, 10 μm scale bar. JEOL JSM 6500F, sample was coated with ~10 nm Au and imaged at 5-15 kV in a vacuum.

Prior to coating with DOPS and cholesterol, the Ti₆Al₄V samples were pretreated in a calcium (Ca) salt solution. Ca is used to bind the DOPS to itself and to the titanium oxide on the surface of the alloy[7]. Controls included samples without the calcium pretreatment, and uncoated Ti₆Al₄V samples.

Scanning White Light Interferometry

SWLI (New View 7300, Zygo Corporation, Connecticut, USA) was used to measure the surface roughness. A 100X objective and 0.5X zoom was used to capture a 140x110 µm field of view with a lateral pixel resolution of 0.22 µm. A composite stitch of 35 of these capture frames was used to analyze SRa and SRz in an area of 0.5x0.5mm. A low pass noise filter was used along with data fill in the analysis of the composite stitch area.

On each coated sample, three 0.5x0.5 mm composite stitched SWLI capture areas were analyzed. Six samples per treatment group were measured. The underlying manufacturing marks were oriented East to West, and the composite capture areas were taken in the center of the sample, 2 mm North of center, and 2 mm South of the center.

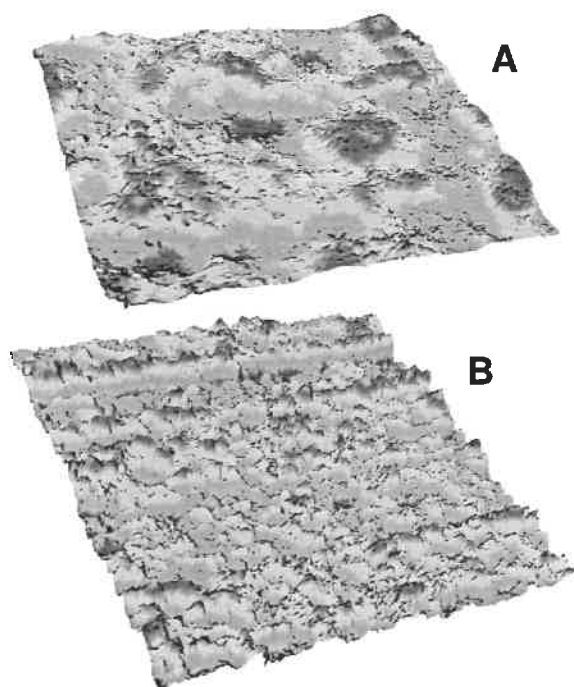


FIGURE 2. Example of a 3D model map captured using SWLI of an individual capture frame (A) and a composite stitched area (B) of a DOPS coated surface. The coating displayed is 100 percent DOPS, with the Ca pretreatment.

Contact Angle Measurements

Aqueous contact angle measurements were used to determine coating hydrophilicity. Contact angle results were taken with a goniometer using 2 µL droplets of de-ionized water. The contact angle of the droplet was measured 5 seconds after initial contact with the coating. Six samples per treatment group were measured.

RESULTS

An example SWLI surface map result for an E-sprayed DOPS and cholesterol surface coating can be seen in Figure 2. Map A from Figure 2 shows a single capture frame (with an area of 140x110 µm), while map B shows a composite stitch area from 35 individual frames and from which SRa and SRz values were calculated (area of 0.5 mm by 0.5 mm).

Contact Angle Results

The contact angle measurement results can be found in Figure 3. The contact angles were lowest (i.e., the surfaces were most hydrophilic) in the 1:1 group (mean = 10.8 degrees). The overall trend was decreasing contact angle with increasing amount of cholesterol in the coatings, with all samples exhibiting relatively low contact angles (below about 20 degrees).

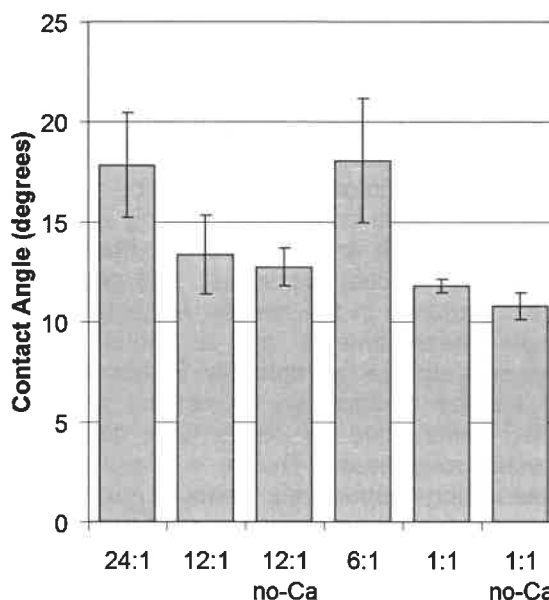


FIGURE 3. Contact angle measurements for the DOPS coated samples. All error bars represent the standard error of the mean.

Roughness Results from SWLI

E-sprayed DOPS and cholesterol coatings show an increase in surface roughness (both SRa and

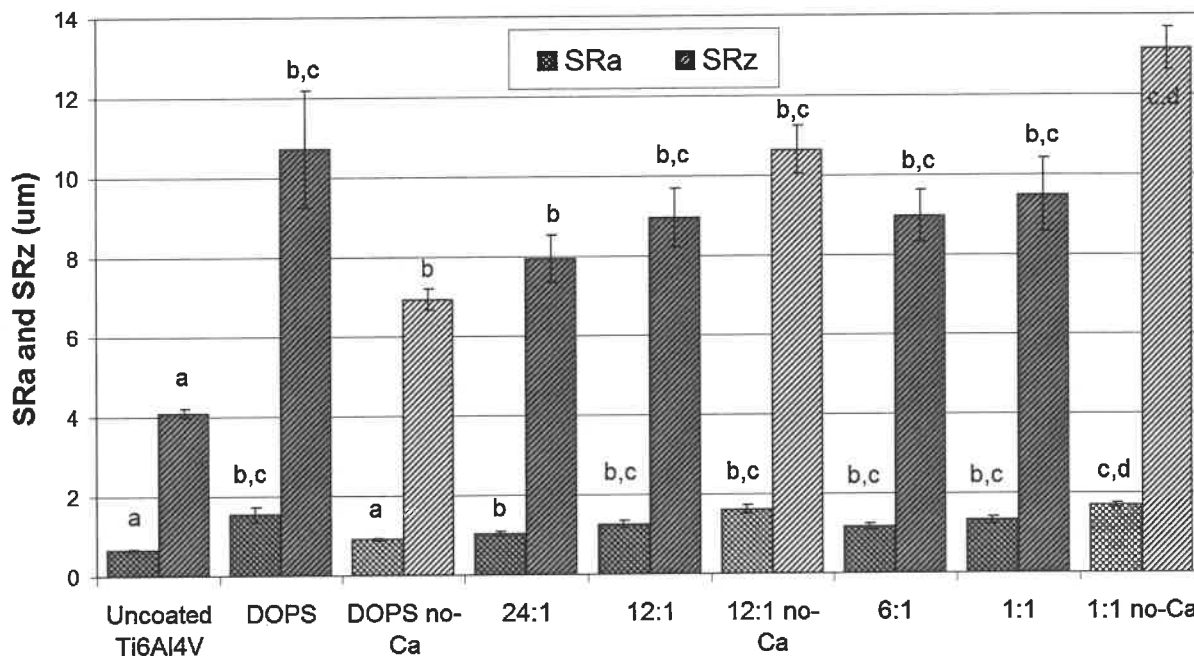


FIGURE 4. Surface roughness (SRa and SRz) of DOPS coated samples. Lighter green and lighter blue bars indicate samples that did not have a Ca pretreatment. All error bars represent the standard error of the mean.

SRz) with increasing concentration of cholesterol (see Figure 4). The most rough coating was the 1:1 group that did not include Ca (SRa mean = 1.69 μm, SRz mean = 13.1 μm). The least rough treatment group was the plain DOPS without Ca (SRa mean = 0.90 μm, SRz mean = 6.9 μm).

Statistical significance between groups is indicated on Figure 4 with letter groups. If two treatments share a letter, they are not significantly different ($p > 0.05$). Results indicate that excluding the Ca pretreatment of Ti₆Al₄V increased the surface roughness in the DOPS and cholesterol coatings while decreasing the roughness in the DOPS-only coating.

Correlation between SWLI and Contact Angle

A Pearson correlation of the means was significant for contact angle and SRa (correlation coefficient = -0.817 with $p = 0.047$) across all compositions. As coating SRa increased, hydrophilicity also increased.

DISCUSSION

The DOPS and cholesterol coatings are semi-transparent and tend to scatter light in such a way as to make measurements of only the top (or bottom) surface of the coating difficult. The use of lower magnification objectives (and their

associated lower numerical apertures) was found to simultaneously collect height information at the coating surface, at anomalies within the coating, as well as from the substrate.

This led to the use of manual control of the scan range in order to physically exclude the substrate surface in the SWLI scan, effectively 'clipping' the lower coating surface. However, this manual 'clipping' technique was not always successful due to roughness of the Ti₆Al₄V substrate being on the same size scale as the thickness of the coatings. Also, manual clipping did not help avoid mid-coating refraction of light or help with coatings that had highly sloped surfaces.

These discoveries led to the use of the 100X objective in order to take advantage of the high numerical aperture and increased ability of capturing only the top surface of the coating. The 0.5X zoom and composite stitch strategy were then utilized in order to increase the analysis area of the coating surface for SRa and SRz measurements. Using these methods, all of the stitched capture areas generally had much greater than 80% height information before data fill, and any missing data was usually only in the high slope regions of the coating.

Correlation of SWLI results to Contact Angle

Biological response to a surface is sensitive to many traits, and surface hydrophilicity is an important characteristic. Contact angle is a useful measure of overall hydrophilicity, but the measurement of surface roughness is essential in order to decouple hydrophilic/hydrophobic effects from chemical composition and/or surface morphology. These results point out that surface roughness, as measured by SWLI, is strongly correlated with contact angle and indicates that surface roughness appears to have more of an effect on contact angle than did coating composition. SWLI is a helpful method for measuring surface morphology and roughness, and has proved useful for investigating effects from coating composition on these DOPS implant coatings.

CONCLUSION

SWLI is an excellent surface analysis technique for characterizing biomimetic DOPS implant coatings. Use of SWLI to capture surface morphology and roughness information of these implant coatings is uniquely beneficial in several ways. First, it allows non-destructive measurement of the coatings. Unlike SEM, which requires a conductive gold (or similar) coating on the surface and decreased atmospheric pressure for imaging, SWLI can be used to measure these coatings before or after other testing is performed.

Second, SWLI measurements are a more repeatable and consistent way to analyze these implant coatings compared to traditional *in vitro* techniques such as contact angle measurements. Contact angle tests have a high amount of inherent variance. With a significant correlation between the SWLI measured SRa and contact angle, SWLI was used to help understand similar aspects of the coating's properties with higher repeatability and precision.

Thirdly, SWLI allows for higher throughput and is less resource intensive than many *in vitro* techniques. This allows for a greater sampling ability and a resulting better understanding of developmental issues in the manufacture of the coatings. Process control issues that were not apparent from small sample size testing in SEM analysis of these coatings have been elucidated in the investigations with the SWLI instrument.

SWLI is a useful tool for the characterization and ongoing development DOPS implant coatings. SWLI was particularly practical for these many-micron thick and rough biomimetic implant coatings in comparison to traditional thin film techniques that are limited to sub-micron coating thicknesses. In the presented work, SRa measured using SWLI was found to significantly correlate with coating contact angle results. SWLI measurements offer the ability to test in series with other *in vitro* measurements as it is non-destructive to the coating. It is a quick, highly repeatable and consistent method.

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THE NEW ISO 10360-7:2011

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INTRODUCTION

An important new international standard for the precision engineering community was published in June 2011. To address the need for common testing methods and specifications for certain types of non-contacting measuring instruments, the working group on coordinate measuring machines (CMMs) within the ISO technical committee 213 developed ISO 10360-7:2011, *Geometrical product specifications (GPS) – Acceptance and reverification tests for coordinate measuring machines (CMM) – Part 7: CMMs equipped with imaging probing systems* [1]. This paper provides an overview of this new standard along with some insight into the options available in the standard. This author was the task force leader for the development of this standard, but the content of this paper should not be considered as any type of official interpretation of the standard.

ISO 10360 SERIES OF STANDARDS

Performance testing standards for CMMs began in 1985 with the publication of ASME/ANSI B89.1.12 [2], and this was soon followed by related standards published by various groups and standards bodies around the world. The historical evolution of CMM testing standards is shown in Figure 1.

The first international standard for CMM testing, ISO 10360-2, was published in 1994 [3]. The solidification and harmonization of CMM testing concepts around the world lead to the publication of a new version of ISO 10360-2 in 2009 [4]. This latest version contains many mature ideas that go beyond any prior national or international CMM test standards.

Until the publication of ISO 10360-7, all the CMM testing standards (such as those shown in Figure 1), only applied to CMMs with contacting probing systems. The goal of ISO 10360-7 was to extend the contacting probing testing ideas to a certain class of CMMs with non-contacting probes. ISO 10360-7 was developed from ISO 10360-2:2009 and contains many of the same concepts and much of the same language.

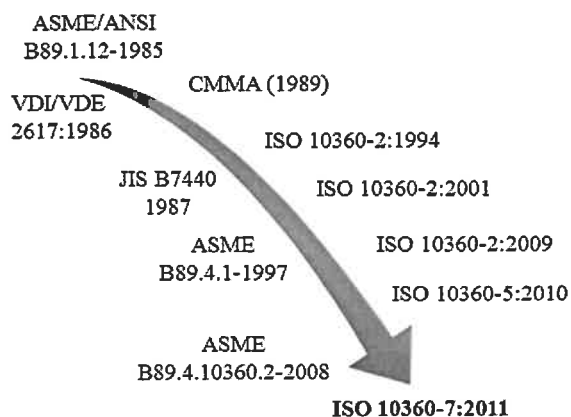


FIGURE 1. The evolution of national and international standards for testing CMMs.

SCOPE OF ISO 10360-7:2011

There are many types of CMMs with non-contact probing systems. ISO 10360-7 only applies to Cartesian CMMs using any type of imaging probing system and operating in the discrete point probing mode. In the metrology market, the largest class of these instruments is the vision or video measuring instrument. An example of an imaging probing system is shown in Figure 2, where the two-dimensional image is nominally parallel to the XY plane of the CMM.

The scope of 10360-7 was specifically chosen based on the market size of CMMs equipped with imaging probing systems plus the lack of existing standardized testing methods. Other types of non-contacting probing systems are being addressed in other parts of the ISO 10360 series of standards, for example see [5].

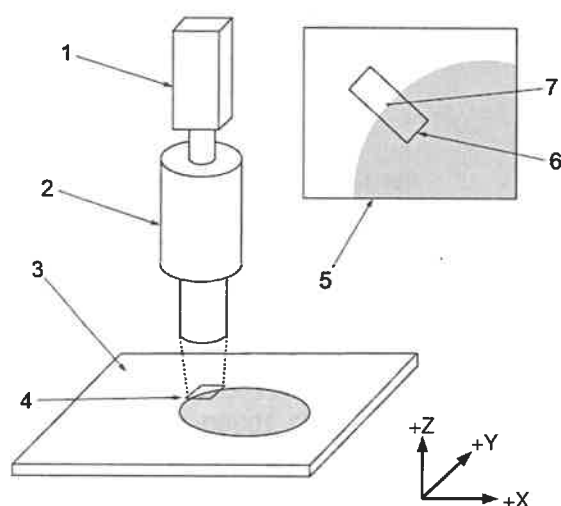
LENGTH TESTS IN ISO 10360-7

ISO 10360-7 offers two alternative methods and specifications for length tests: the *composite* approach and the *component* approach.

Composite Approach

The composite approach adapts the exact same length testing approach as used for contacting probe CMMs in ISO 10360-2:2009. In 10360-2, the E_0 test involves measuring a step gage (or similar artifact) along seven measurement lines

in the volume of the CMM, with the four spatial diagonals being required, and the other three lines usually being parallel to the machine axes. Along each measurement line, five different lengths are measured three times each, for a total of 105 measurement values. The 105 E_0 values are compared to $E_{0,MPE}$, the maximum permissible error (MPE) of the CMM under test, to determine conformance. To allow for consistency with contacting probe CMMs, the composite approach in 10360-7 applies this same testing method to CMMs equipped with imaging probing systems.



Key

- 1 device for capturing an image of the measured object
- 2 various optical elements of the imaging probing system
- 3 measured object
- 4 field of view (object)
- 5 field of view (image)
- 6 measuring window
- 7 measured point

FIGURE 2. Example imaging probing system.

Bidirectional versus Unidirectional

In comparison to 10360-2, 10360-7 does allow one key difference in regards to the type of artifact being used for the testing. For the E_0 test in 10360-2, the test artifact must be measured in a bidirectional manner, i.e. the two points that are used in the measurement of a length must be in opposing directions. See Figure 3. In 10360-7, both bidirectional and unidirectional measurements are allowed. The notation in 10360-7 is therefore slightly different than 10360-2; E_B and E_U are used to indicate bidirectional or unidirectional testing to the corresponding $E_{B,MPE}$ or $E_{U,MPE}$ specifications. The choice of specification is at the discretion of

the CMM manufacturer, and 10360-7 does not recommend one approach over the other.

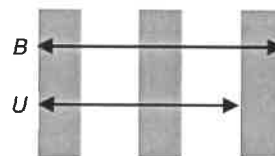


FIGURE 3. Many artifacts, e.g. a linescale, allow both unidirectional measurements (U) and bidirectional measurements (B). The artifact must be calibrated for the desired use.

Component Approach

The component approach is based on common industrial testing methods for imaging probe CMMs and includes three separate tests. The first test is a modification of the 10360-2 E_0 test applied to the XY plane, E_{UXY} or E_{BXY} , and is tested along four measurement lines parallel to the XY plane with a linescale or similar artifact (see Figure 4). The second test involves length measurements parallel to the Z axis, E_{UZ} or E_{BZ} , and is tested along one measurement line with gage blocks or similar artifacts (see Figure 5).

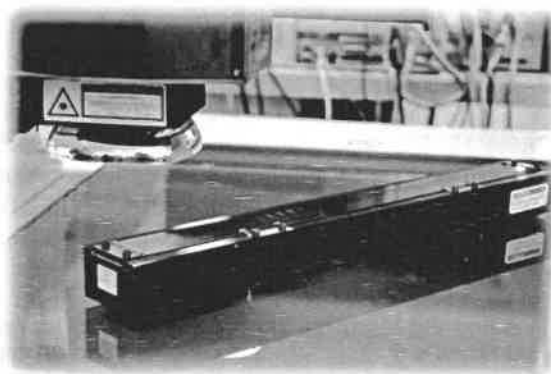


FIGURE 4. Example testing of E_{UXY} along one of two required planar face diagonals. The default for the other two positions is parallel to the axes.

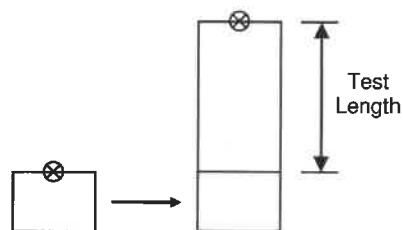


FIGURE 5. Example test of E_{UZ} using one gage block stacked onto another one which is used as the reference point.

Z-axis Motion Relative to X and Y Axes

A major goal of 10360-7 was to ensure that imaging probe CMMs have three-dimensional specifications. Even when these instruments are used for two-dimensional measurements, it is common for the instrument to move in all three dimensions. As shown in Figure 6, when a measurement involves projecting measured features into a common plane, such as is needed to measure the distance between the two bores shown in the figure, the Z axis is typically moved up or down. The error motion of the Z axis relative to the XY plane is often overlooked and can result in large errors in what are usually considered to be just two-dimensional measurements. This error motion is typically dominated by the Z-axis squareness error (to the X and Y axes) and therefore 10360-7 has a Z-axis squareness test, E_{SQ} . An example E_{SQ} test using a square is shown in Figure 7.

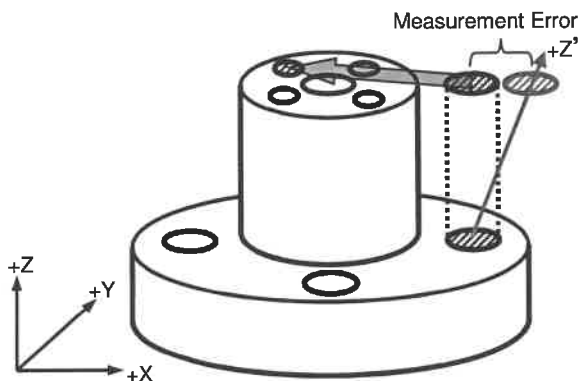


FIGURE 6. Measurement error associated with a squareness error in the XZ-plane. The Z' axis represents a possible error motion in an actual Z-axis, which results in measurement errors in two-dimensional measurements, as shown.

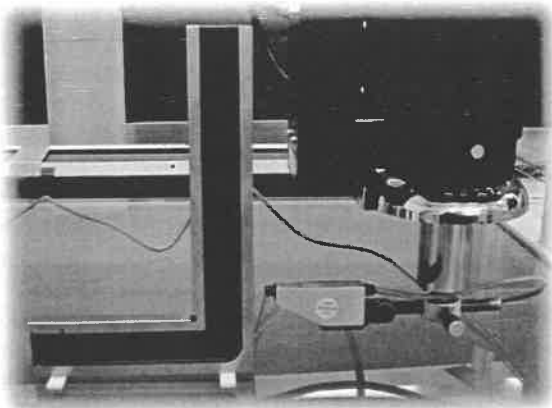


FIGURE 7. Example test of E_{SQ} with a square.

The measurement of a mechanical square for testing E_{SQ} may require the use of an indicator, or indicating gauge head, as shown in Figure 7. This approach has some drawbacks and was carefully considered in the development of the 10360-7 standard. The standard also allows for alternative testing approaches of E_{SQ} .

Composite versus Component Approach

ISO 10360-7 offers two approaches to length testing and does not recommend one approach over the other. The composite approach may be more useful when comparing the performance of imaging probe CMMs to contacting probe CMMs. The component approach may be more useful when comparing new imaging probe CMMs to prior products (which used similar specifications). The component approach may also be more useful when the measurement application is primarily two-dimensional; however, both approaches address the three-dimensional nature of the CMM. In addition, the MPE values associated with the two approaches cannot necessarily be directly compared.

PROBING TESTS IN ISO 10360-7

In addition to length tests, ISO 10360-7 also includes a so-called probing performance test, P_{F2D} , which is based on other probing tests within the ISO 10360 series [6]. For P_{F2D} a test circle is measured using 25 points in a manner such that the CMM moves between all consecutive points and that the measuring windows are distributed across the entire field of view. From the 25 points, a single circle is calculated and the form (maximum minus minimum radial distance) is reported as the P_{F2D} measured value. An example of an allowable measuring pattern for P_{F2D} is shown in Figure 8. The test circle for use in the P_{F2D} test must be calibrated for the form of the circle (i.e. the circle roundness).

TESTING FIELD OF VIEW SEPARATELY

ISO 10360-7 offers two optional tests associated with the measuring performance of the field of view separate from the rest of the CMM. The length test, E_{UV} or E_{BV} , and the probing test, P_{FV2D} , are similar to the tests described above but are done with no motion of the CMM. The test for E_{UV} or E_{BV} is similar to E_{UXY} and E_{BXY} respectively but requires the four measurement lines to be within the field of view. Similarly, the test for P_{FV2D} is identical to that of P_{F2D} but uses a smaller circle that can be completely contained within the field of view. The specification of an

MPE associated with these field of view only tests is at the discretion of the imaging probe CMM manufacturer.

ISO 10360-7 does not provide any recommendation concerning the application of these optional specifications. E_{UV} , E_{BV} , or P_{FV2D} will be more applicable to some imaging probe CMMs than others, and some manufacturers may find these specifications useful to highlight features of their particular instruments.

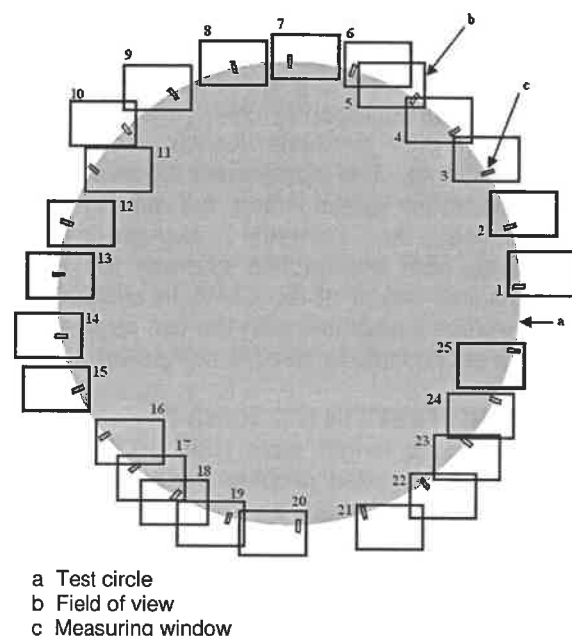


FIGURE 8. Example of an allowable test pattern for measuring P_{F2D} . Note the distribution of the 25 measuring windows across the field of view.

SPECIFICATION TO ISO 10360-7:2011

A summary of the MPE specifications to ISO 10360-7:2011 that would be expected from the imaging probe CMM manufacturers includes:

For the composite approach:

$$E_{B,MPE} \text{ or } E_{U,MPE}$$

$$P_{F2D,MPE}$$

For the component approach:

$$E_{BXY,MPE} \text{ or } E_{UXY,MPE}$$

$$E_{BZ,MPE} \text{ or } E_{UZ,MPE}$$

$$E_{SQ,MPE}$$

$$P_{F2D,MPE}$$

For both approaches, the following are optional:

$$E_{BV,MPE} \text{ or } E_{UV,MPE}$$

$$P_{FV2D,MPE}$$

OPTIONS IN ISO 10360-7:2011

Unlike most of the current test methods in the ISO 10360 series, ISO 10360-7 offers a number of options. These options include the composite versus component approach, the use of either bidirectional or unidirectional artifacts, and the optional field of view only specifications. These options allow for greater acceptance of the new standard in industry, but there is some risk that too many options will lead to incomparable specifications. In the introduction clause of ISO 10360-7, it is mentioned that the differences between ISO 10360-7 and ISO 10360-2 may be eliminated in future revisions of either standard.

SUMMARY

The test methods in the new ISO 10360-7:2011 were presented and some of key options discussed. It is expected that this standard will have a positive impact on the imaging probe CMM market and will significantly benefit users of this technology by bringing about a better understanding of measurement performance.

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Measurement Uncertainty of High Energy Density Science Target Assemblies for the National Ignition Facility

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Introduction

Successful High Energy Density Science (HEDS) experiments at the National Ignition Facility (NIF) [1] require that millimeter-scale targets are fabricated, assembled, and measured with micrometer-scale tolerances. However, the materials and geometries involved in these assemblies present challenges for metrology and uncertainty quantification. The completed assemblies must accommodate both the features required for the physics experiment, as well as facility-dictated attributes such as shields, coatings and alignment fiducials. Here, we present an error analysis of target feature measurements, and propose solutions for minimizing uncertainty in future target designs.

Because of the fragility of the individual components and assemblies, optical metrology techniques [2] are used almost exclusively as a means of qualifying the final product. However, difficult-to-measure components such as dimpled metal shields, plastic coatings, and transparent features introduce uncertainties into these measurements and often require additional test procedures to determine the measurement uncertainty [3]. Furthermore, metrology of some individual components requires the use of alternate measurement techniques (e.g., contact probes). Combining these task-specific measurements [4] introduces additional uncertainties and has direct consequence to the success and/or failure of the experiment in terms of both interpreting the experimental results and properly aligning the target in the NIF target chamber.

Several different HEDS experiments are fielded on NIF, each requiring a different target configuration. The targets described here are all indirect drive experiments, where the laser beams are pointed into the laser entrance hole (LEH) of a gold hohlraum aligned to the target chamber center (TCC). The 3Ω laser energy is converted into x-rays when the beams interact

with the gold on the inside of the hohlraum. This

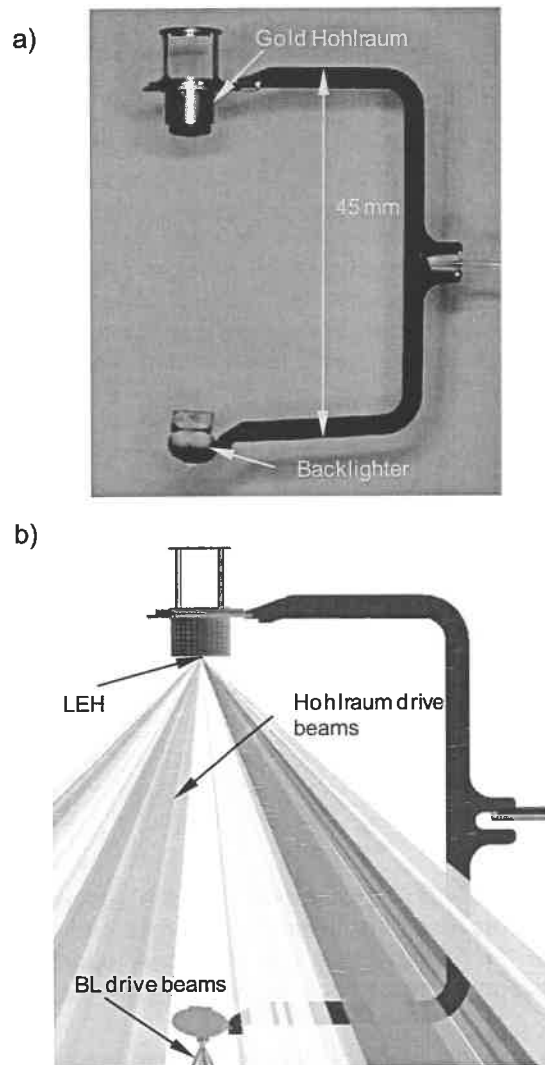


FIGURE 1. a) Photograph of a streaked radiography target assembly. b) Model showing the laser drive beam entering the LEH and the backlighter drive beams hitting the microdot.

is the energy source for the experiment. Figure 1 illustrates one target configuration used to image x-ray flux through a low density material. The laser drive beams are pointed at a backlighter

assembly (BL) directly below the LEH. The backlighter consists of a silver microdot on a Ta substrate with a laser machined slot that acts as the imaging x-ray source for the x-ray camera pointed at an aperture in the upper target assembly.

A schematic representation of the target/beam alignment is shown in Figure 2. The Target Alignment System (TAS) is located within the target chamber relative to the Chamber Coordinate Reference System (CCRS). Subsequently, the laser beams and Target Positioner (TARPOS) are aligned to the TAS. The positional accuracy required for a target relative to the TAS is $\pm 25 \mu\text{m}$ over the 5.0 m radius of the target chamber.

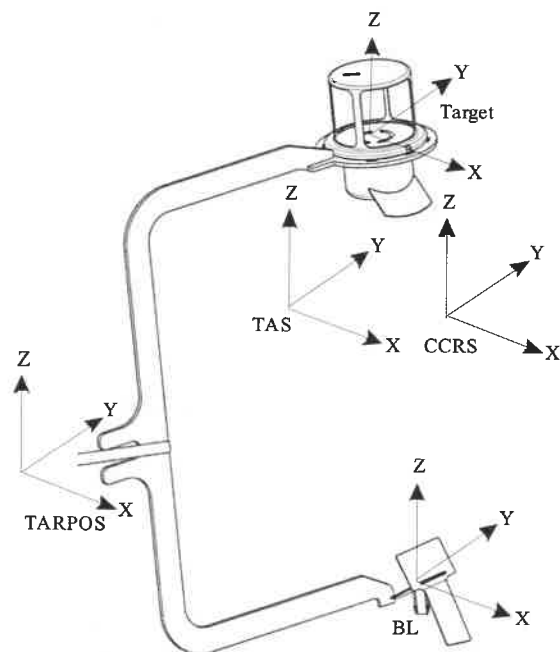


FIGURE 2. Illustration of the coordinate systems used for aligning a target within the NIF. Not shown are the additional coordinate systems representing diagnostic alignment.

In addition, various diagnostics are aligned to the target based on the location of TCC as determined by the TAS. The alignment accuracy of the diagnostics varies based on the type of system being used. For x-ray imaging systems, the allowable alignment error is on the order of $\pm 500 \mu\text{m}$. For systems like the Velocity Interferometer System for Any Reflector (VISAR) used for measuring shock velocity, $\pm 100 \mu\text{m}$ can be required. The goal of this work is to quantify the uncertainty in the location of the

target relative to TCC, based on measurement and alignment uncertainties.

Analysis

After target fabrication is complete, and before the target is transferred to the NIF target chamber, metrology of the target assembly is performed with optical coordinate measuring machines (OCMMs). To quantify the global effect of the task-specific measurement uncertainty, homogenous transformation matrices (HTMs) assuming small angles [6] were used to model the error sources and relate their influence to the overall target installation error budget. As is common in coordinate metrology, a local part or subassembly coordinate system (CS) is created. In the example shown in Figure 2, two separate coordinate systems will be defined for the target package and backlighter assembly.

A point derived from the OCMM is represented as follows

$${}^{OCMM}\{P\} = \begin{Bmatrix} P_x + \delta x \\ P_y + \delta y \\ P_z + \delta z \end{Bmatrix}, \quad (1)$$

where P_x , P_y , and P_z are the X, Y, and Z coordinates and δx , δy , and δz are the measurement errors for each coordinate respectively. Each point is related to the part/subassembly CS expressed by

$${}^{CS}\{P\}_{OCMM} = {}^{CS}[T]_{OCMM} {}^{OCMM}\{P\}, \quad (2)$$

where $[T]$ is a transformation done by the OCMM. These points are used to create geometric features representing the measured assembly. These geometric features include the task specific measurement uncertainty which consist of errors associated with the machine, probe, algorithms, edge quality, the influence of coatings, etc.

Once each of the components and/or subassembly is defined by a set of points or geometric features the subassemblies are related to a common reference. For the case shown in Figure 2 the BL and TARPOS are defined relative to the target CS by

$${}^{Tar}\{P\}_{BL} = {}^{Tar}[T]_{BL} {}^{BL}\{P\}, \quad (3)$$

$$^{Tar}\{P\}_{TPos} = ^{Tar}[T]_{TPos} ^{TPos}\{P\}. \quad (4)$$

Now the target is completely defined relative to the target CS. The final transformation relates the target assembly in the TARPOS to the TAS

$$^{TAS}\{P\}_{Tar} = ^{TAS}[T]_{Tar} ^{Tar}\{P\}, \quad (5)$$

where

$$^{Tar}\{P\} = ^{Tar}\{P\} + ^{Tar}\{P\}_{BL} + ^{Tar}\{P\}_{TPos}. \quad (6)$$

The error between the measured features and the TAS calculated by

$$^{TAS}\{E\} = ^{TAS}\{P\}_{Ideal} - ^{TAS}\{P\}_{Measured}. \quad (7)$$

Removing second order terms, the error $\{E\}$ for each coordinate is given by

$$\begin{aligned} ^{TAS}E_X &= \varepsilon_{y,TAS}(\sum Z_{offsets}) - \varepsilon_{y,TAS}(\sum Y_{offsets}) \\ &+ \varepsilon_{y,TPos}(P_{z,TPos}) + \varepsilon_{y,BL}(P_{z,BL}) - \varepsilon_{z,TPos}(P_{y,TPos}) \\ &- \varepsilon_{z,TPos}(P_{y,BL}) + \sum \delta_{x,comp}, \end{aligned} \quad (8)$$

$$\begin{aligned} ^{TAS}E_Y &= \varepsilon_{z,TAS}(\sum X_{offsets}) - \varepsilon_{x,TAS}(\sum Z_{offsets}) \\ &+ \varepsilon_{z,TPos}(P_{x,TPos}) + \varepsilon_{z,BL}(P_{x,BL}) - \varepsilon_{x,TPos}(P_{z,TPos}) \\ &- \varepsilon_{x,TPos}(P_{z,BL}) + \sum \delta_{y,comp}, \end{aligned} \quad (9)$$

$$\begin{aligned} ^{TAS}E_Z &= \varepsilon_{x,TAS}(\sum Y_{offsets}) - \varepsilon_{y,TAS}(\sum X_{offsets}) \\ &+ \varepsilon_{x,TPos}(P_{y,TPos}) + \varepsilon_{x,BL}(P_{y,BL}) - \varepsilon_{y,TPos}(P_{x,TPos}) \\ &- \varepsilon_{y,BL}(P_{x,BL}) + \sum \delta_{z,comp}, \end{aligned} \quad (10)$$

where X, Y, and Z are relative offsets between features in the TAS CS, ε_x , ε_y , and ε_z are the angular errors between CSs and δ_x , δ_y , and δ_z , are the displacement errors between CSs.

The analysis shown above represents key aspects of target manufacturing, with respect to metrology and system integration. The first two terms of eqs 8-10 represent the influence of the angular error and feature offsets between the target and the TAS within the NIF target chamber. In addition, the second set of terms in eqs 8-10 represent the tolerances required to define critical features, such as datums used to relate the BL and TARPOS to the target. This represents how well features need to be measured, but more importantly, how well datum features need to be defined. For the example shown in Figure 2 the calculated tolerance

between the BL and Target and TARPOS and Target is on the order of 25 μm , which would ideally be verified by metrology having a measurement uncertainty of on the order of 2.5 μm .

Given these stringent requirements for alignment in the NIF chamber, it is necessary to integrate features specifically fabricated for this purpose into target designs. Depending on the type of experiment and drive configuration being utilized, there have been a number of different methods used to define alignment features. Figure 3 shows three different methods of transferring alignment information in the NIF target chamber. The target shown in a) has a 300 μm thick silicon washer as part of the target structure that acts as datum for both target metrology and alignment within the chamber. The Si washer has a total thickness variation of less than $\pm 1 \mu\text{m}$ and a flatness of better than $\pm 0.05 \mu\text{m}$ over the 9 mm diameter.

However, integrating a quality reference feature is not always applicable as illustrated by the examples shown in Figure 3 b) and c). In these cases, auxiliary parts such as dimpled aluminum shields used to reflect stray unconverted light from interfering with diagnostics impede measurement quality and often require additional features to be added specifically for chamber alignment. To transfer the datum information for alignment in the target chamber external fiducials have been added and measured on an OCMM to allow for insertion to a positional accuracy on the order of 25 μm .

Summary

Target alignment within the NIF chamber is a critical part of the overall success of an experimental campaign. Due to the stringent alignment requirements, target production including part/subassembly metrology and datum derivations needs to be considered throughout the design process. Typical measurement uncertainties for critical features is less than 10 μm , but can vary based on the overall target design and types of diagnostics being utilized during the experiment. In addition to alignment accuracy it is also important to have repeatable alignment procedures to expedite the shot cycle.

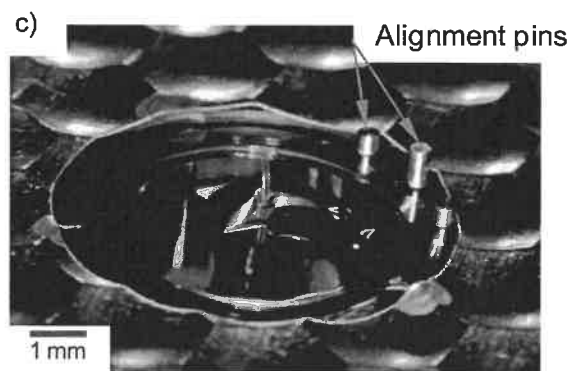
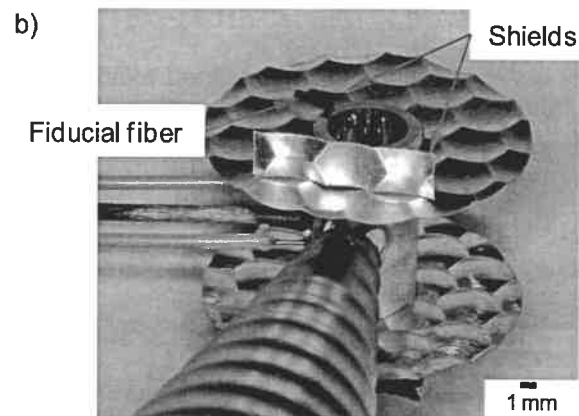
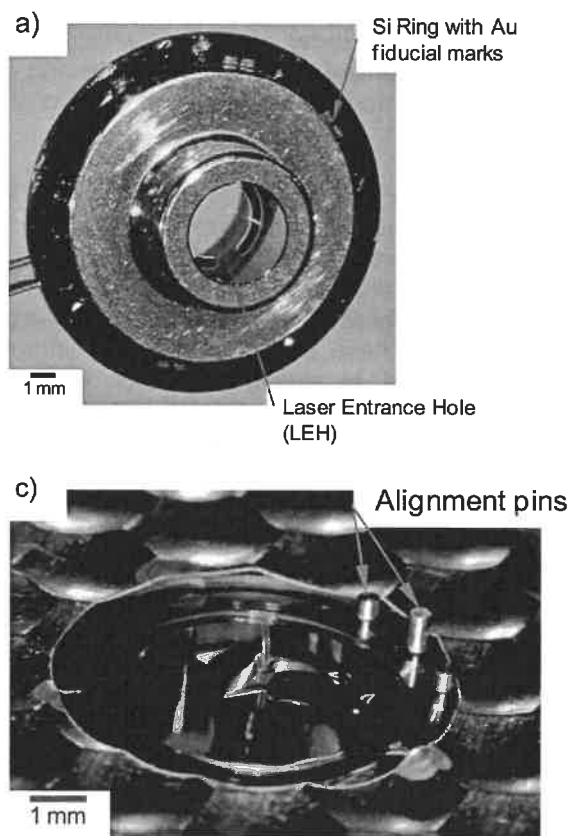


FIGURE 3. a) HEDS single sided drive target configuration using Si ring with gold fiducial marks to create datum for both metrology and alignment in the NIF chamber. b) Equation-of-state target with alignment fiducial for setting Z location in the NIF target chamber. c) Alignment pins used to determine angular orientation and Z position within the target chamber.

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Thickness and Stack Height Measurement Uncertainty of Experimental Packages for National Ignition Facility Targets

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Introduction

Targets for experiments carried out at the National Ignition Facility (NIF) [1] often incorporate millimeter-scale stacks of various materials with individual thicknesses small as a few micrometers. Thickness measurements of the individual components, as well as any included glue lines or gaps, are essential for successful experiments and data analysis. Part geometries and materials necessitate multiple measurement techniques that must be combined to infer micrometer-scale glue thicknesses from stack height measurements. Furthermore, form variation in thin parts makes quantifying errors challenging. Here, we discuss issues involved in the metrology and uncertainty quantification for these types of experimental packages.

Target materials of interest include low density foams, metal foils and transparent crystalline solids. Typically, the components contain precision-manufactured features such as steps, slots or sine waves [2]. Multiple components are assembled in a stacked geometry such as the arrangement illustrated in Figure 1. Tolerances for these parts and assemblies are often at, or below, the micrometer level for feature size, thickness and form. These tolerances are similar to those required for MEMs applications [3].

For many targets, the stacks are assembled with micrometer-scale glue layers, which are too thin to be measured directly with techniques such as optical or x-ray microscopy. Furthermore, the geometry of the assembled parts obscures the glue layer. While the individual stack components can be readily measured with tools such as measuring microscopes and touch probes, metrology of the assembled stack presents challenges because glue layer thickness and uniformity must be inferred from indirect measurements.

Measurements and Methods

In general, both contact and non-contact metrology methods are used in combination to evaluate the individual parts and assemblies at various stages of the assembly process. A variety of instruments including "on-machine" gauging techniques are used throughout the manufacturing process. In many cases, an individual component is measured most accurately using one technique before assembly, but after assembly that measurement technique cannot be used. For example, the LiF and Ta foil shown in Figure 1 are measured with a contact probe before assembly, but after the

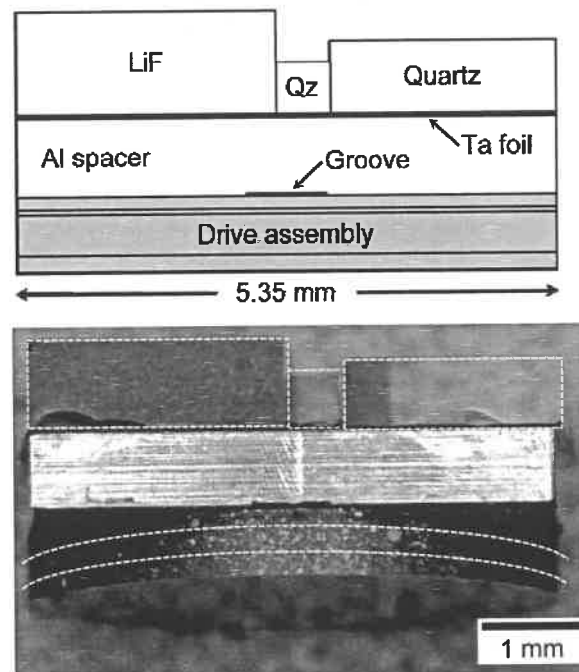


FIGURE 1. Scale drawing of experimental package and photo of assembly shown above. The LiF and quartz (Qz) windows are transparent. The drive assembly consists of two layers of CH, and two densities of carbon foam. These layers are encapsulated in epoxy, and appear as one layer in the photograph.

LiF is glued to the Ta, the stack is too delicate for a contact measurement. In this case, the

stack height is measured using white light interferometry (WLI), and the glue thickness is inferred. The variation in measurement uncertainty between contact and non-contact metrology for the large variety of materials used in an experimental package adds directly to the overall measurement uncertainty and represents the focus of this work.

Typically, thickness and form measurements are done by placing the part or assembly onto a reference surface, typically a gage block or optical flat, and then using a contact or non-contact probe technique to relate features of interest on the part to the reference surface. Common technologies used include white light interferometry, laser probes, and both uni-directional and multi-directional contact probes.

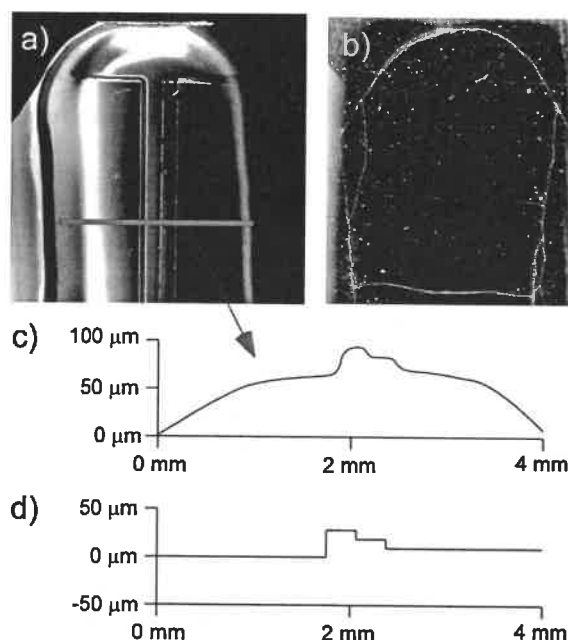


FIGURE 2. a) Photograph of Ta steps coated onto a polycrystalline carbon substrate. b) Backside photograph of the Ta coating showing cracking on the periphery. c) Lineout across the steps of the Ta part (data smoothed) d) Illustration of the ideal lineout.

Figure 2 illustrates the types of features and form error that need to be characterized as part of the overall assembly. In this case, tantalum steps have been sputter-deposited onto a polycrystalline carbon substrate. Figure 2 c) shows a lineout of the measured sample. Figure 2 d) is the ideal profile based on the design. The substrate has warped and cracked due to residual stress in the film resulting from the coating process (Figure 2 b). In this case there is approximately 100 μm of form on a nominally 100 μm thick sample. The ability to deconvolve

thickness variation or gaps from form error is a key requirement for understanding the experimental data.

Other methods of making these types of measurements include x-ray computed tomography (CT) and opposing probe techniques. [4, 5] However, these techniques are beyond the scope of this effort.

Analysis

To quantify the task-specific measurement uncertainty involved in measurements where individual components are measured with a different technique than their resulting stack heights, three grade 00 gage blocks with an expanding uncertainty of 60 nm were wrung together to create a stepped sample, as shown in Figure 3. This sample was then measured by two different WLIs, an optical coordinate measuring machine (OCMM) with a laser probe, and an automated contact-based length gage. Table 1 lists the manufacturer's instrument specification for each tool used in this study.

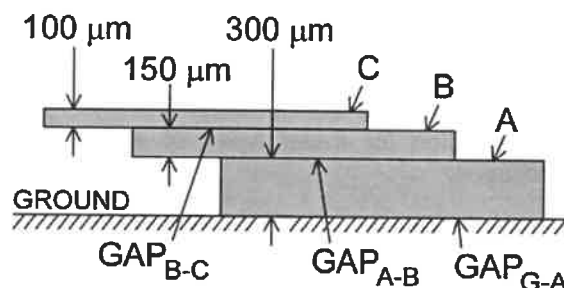


FIGURE 3. 300 μm , 150 μm and 100 μm grade 00 gage blocks wrung together and placed on a ground steel substrate. The reference surface (ground) has a surface finish of 10 nm R_a and a flatness of 50 nm.

Each step is measured relative to the reference surface using each of the instruments listed previously. The gage length, including errors associated with the gage block artifact measurements, can be expressed by

$$L_{GA} = L_A + \delta_A + \delta_{GA}, \quad (1)$$

$$L_{GB} = L_{GA} + L_B + \delta_B + \delta_{AB}, \quad (2)$$

$$L_{GC} = L_{GB} + L_C + \delta_C + \delta_{BC}, \quad (3)$$

where L_A , L_B , and L_C are the accepted lengths of the gage blocks, δ_A , δ_B , and δ_C are the gage block errors and δ_{GA} , δ_{AB} , and δ_{BC} represent the errors/gaps between the reference surface (ground) and gage A, gage A and gage B and gage B and gage C respectively. Removing the accepted gage block length, the errors associated with the artifact are given by

$$E_{GC} = \delta_A + \delta_{GA} + \delta_B + \delta_{AB} + \delta_C + \delta_{BC}. \quad (4)$$

For a stack with n materials and n interfaces (including the interface with the reference surface), the general form of Equation (4) is:

$$E_{0,n} = \sum_{i=1}^n \delta_i + \sum_{i=0}^n \delta_{i,i+1}, \quad (5)$$

where δ_i is the error associated with each individual part in a stack and $\delta_{i,i+1}$ is the error/gap of each interface in the assembly stack of interest.

For the analysis presented here, δ_A , δ_B , and δ_C are the expanded uncertainties of each of the three gage blocks given by the calibration certificate as $(0.06+0.5 \cdot L/1000) \mu\text{m}$ where L is in mm. The surface finish and flatness of the gage blocks used were less than 10 nm R_a and 50 nm, respectively. Considering the lengths of the gages in use, 60 nm was used as the gage expanded length uncertainty.

TABLE 1: Instrument specifications used for single-sided thickness measurements.

Instrument	Manufacturer Specification	Range (mm)	Probe Type
OCMM	$1.5+L/150 \mu\text{m}$ (L in mm)	250	Laser
Length Gage	$1 \mu\text{m}$ (0.5 μm res)	12	Stylus
WLI (1)	$0.025 \mu\text{m}$ (over 500 μm range)	8	Interferometer
WLI (2)	$0.025 \mu\text{m}$ (over 500 μm range)	12	Interferometer

Results

The gage block sample was measured ten times using each of the four measurement systems described in Table 1. The sample was removed and replaced for each trial. Each measurement was done in a temperature controlled class 1000 clean room with a ΔT of less than $\pm 1.0 \text{ C}^\circ$. Using a thermal expansion coefficient of $11.5 \text{ e-}6 \text{ m/m}^\circ\text{C}$ [6] the total variation of the gage block length due to temperature variations is approximately 12 nm and considered negligible for this comparison.

The interface errors were solved using eqs (1-3) and assuming the expanded uncertainty of the gage block thickness to be 60 nm. It was expected that the interface errors between gages would be consistent with those commonly found in literature, which have been reported to range anywhere from 6 nm to 50 nm [7].

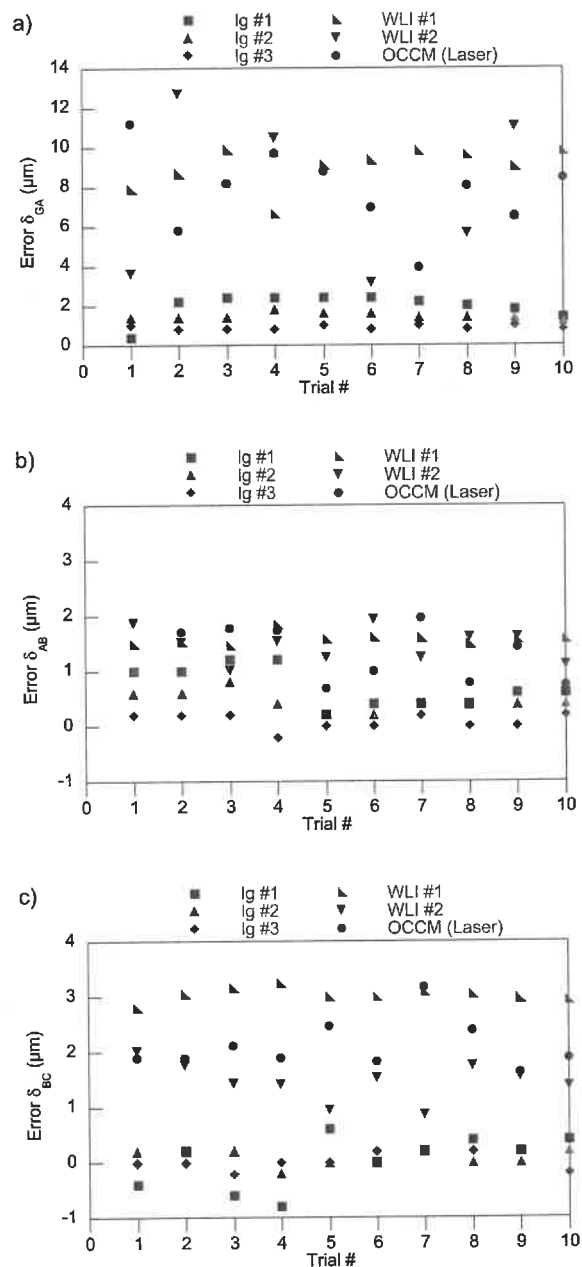


FIGURE 4. Plots of the measured gap errors a) Error between the reference surface and gage block A. b) Error between gage A and gage B. c) Error between gage B and gage C.

Figure 4 shows the results of interface error for ten trials from each instrument. The automated length gage was operated using three different trigger settings, corresponding to increasing probe pressure (lg#1, lg#2, lg#3). It was clear from the data that the error between the sample and the ground or reference surface was large compared to the relative gap between gage blocks, as expected. However the errors or gaps

between gage blocks were larger than expected for wrung gages, and they represent the types of gaps or errors that need to be quantified to the micrometer level for NIF targets.

The average and standard deviation for each gap measurement are shown in Table 2. Clearly the contact-based measurements lowered the influence of the gaps, primarily between the reference surface and the first gage. The error in stack heights resulting from the instrument and/or measurement technique is on the order of glue thicknesses and gaps measured in target assemblies. As a result, stack height measurements may or may not provide the information required to relate pre-experiment characterization to the experimental data.

TABLE 2. Average and standard deviations of the gap measurements.

Instrument	δ_{GA} (μm)		δ_{AB} (μm)		δ_{BC} (μm)	
	Ave	σ	Ave	σ	Ave	σ
Gage (#1)	2.0	0.6	0.7	0.4	0.02	0.5
Gage (#2)	1.5	0.2	0.5	0.2	0.1	0.1
Gage (#3)	0.9	0.1	0.1	0.1	0.04	0.1
WLI (1)	8.98	1.02	1.57	0.11	3.03	0.12
WLI (2)	12.25	6.69	1.47	0.31	1.47	0.35
OCMM (laser)	7.8	2.0	1.3	0.5	2.0	0.5

Future Work

The authors are currently developing appropriate measurement artifacts to evaluate the performance of the optical-based systems when they are used to measure multiple reflective surfaces, such as quartz or lithium fluoride (LiF). Double-sided thickness measurements can alleviate many of the stack height errors associated with single-sided measurements. A tool for making these measurements with errors of $< 1 \mu\text{m}$ is currently being developed (see reference [5]).

Summary

As expected, there is a relatively large error (4 μm to 12 μm for non-contact measurements and 1 μm to 3 μm for contact measurements) between a reference surface and a gage block sample. The contact probe techniques reduce the gap error to within the limits of instrument accuracy. However, contact probes are often unacceptable for delicate target assemblies. Contact stress can crack, scratch, or plastically deform the part during the measurement process. In addition, the influence of the probe

on the part can produce values not representative of the assembly in its "free" state. NIF experiments are sensitive to micrometer variations in thickness, so introducing perturbations on this scale during characterization can affect the results of the laser experiments.

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Simulation and Experimental Investigation on Radial Run-out Error of Rotary Air Bearing Tables

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ABSTRACT

In this paper, a transfer function method for predicting radial motion error of rotational axis is presented and experimental assessment of the simulation is also presented. In order to simulate the radial error motion generally full discretized equation should be solved which consumes much calculation time. The transfer function method suggests that the error motion due to a segment of a bearing can represent the total whole bearing error motion characteristic.

INTRODUCTION

Figure 1 shows the whole flow chart of our simulation platform for rotary system and eventually this platform simulator is included in the total machine (such as machine tools or multi-axis stages for LCD manufacture) simulation platform. This simulation platform deals with various bearing types and this paper describes a part of air bearing type rotary system simulator.

For many precision rotary stage applications the air bearing guide is adopted because of no friction motion, no heat generation and good run-out performance. In order to design the air bearing stage the air flow model in the bearing clearance gap and restrictor should be made. And generally those physical models are solved by numerical method. By using that method bearing physical characteristics such as stiffness and damping can be obtained and with the machining error (bearing and rotor surface) and the rotation eccentricity the radial error can also be obtained. In order to obtain radial error during one rotation, at every small angle step the full numerical calculation should be done and therefore much calculation time is needed.

Transfer function method was proposed to solve this problem. In this method the transfer function has physical quantity of the exerting force on the rotor at each angle and the transfer function

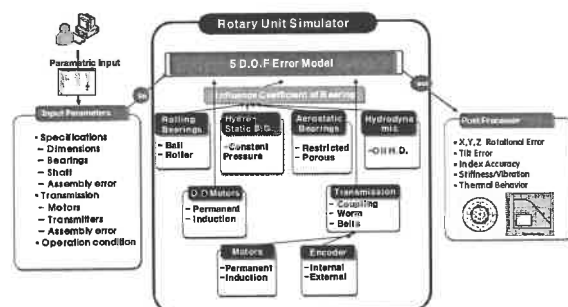


FIGURE 1. Simulation structure of the total rotary system simulating platform.

value is calculated for each spatial frequency value. The spatial frequency means that the spatial frequency of machining error around the bearing surface circular path. And the stiffness of the bearing can be obtained by numerical simulation and therefore, with the transfer function just like Fourier Transform components, any form of surface machining error (radial direction) effect on the run-out error can be simulated by integration of each product of the surface error and the transfer function. For numerical simulation FDM (Finite Difference Method) was used and Reynolds equation was solved by ADI (Alternating-Direction Implicit) method. In this paper the comparison between full numerical calculation by FDM and the proposed transfer function method is shown and investigation on the usefulness and effectiveness of the transfer function method is discussed. And an experimental investigation on assessing effectiveness of those simulation methods will be presented.

MODELING AND SIMULATION

Figure 2 shows the simple journal air bearing schematic. Feedholes are located around the journal and the number of feedhole layer is normally from one to two. In order to simulate air bearing characteristic Reynolds equation should be solved. Neglecting shaft radial direction component terms, two-dimensional Reynolds equation can be obtained. The Reynolds

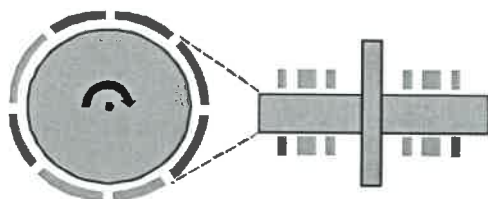


FIGURE 2. Schematic view of air bearing journal

equation is generally solved by numerical method such as FDM. The ADI method is adopted for calculation speed and algorithm versatility. Figure 3 shows the simulation results which has two feedhole layers and each layer has eight feedholes. With the formulated simulation algorithm, error motions of rotational air bearing have been simulated. The journal surface was assumed to have form errors and therefore when the table rotates to an angle the changed boundary condition causes pressure

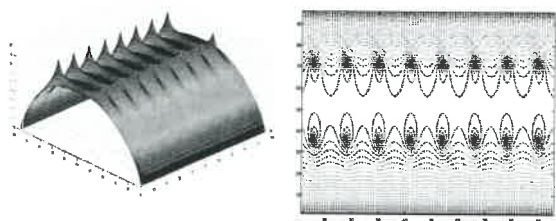


FIGURE 3. Air bearing pressure distribution results from FDM simulation.

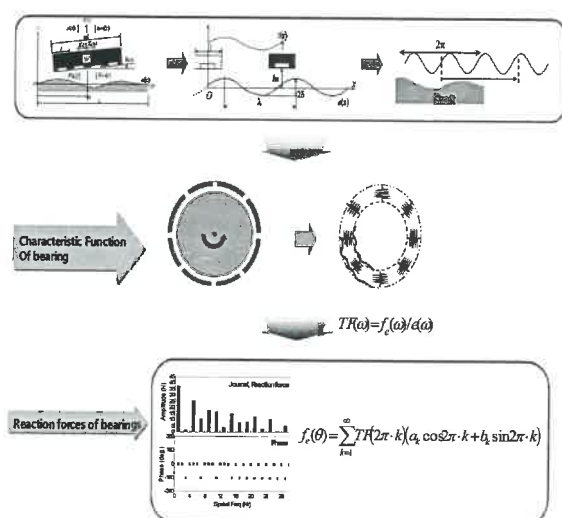


FIGURE 4. A schematic concept of transfer function method for simulating rotational motion.

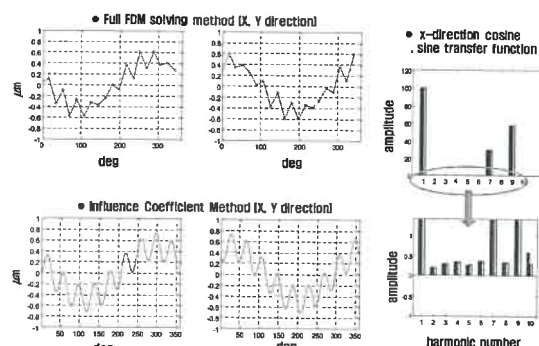
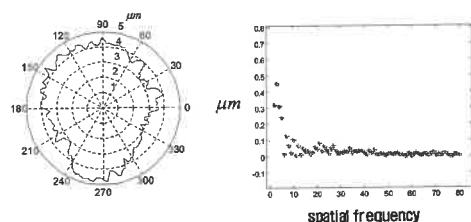


FIGURE 5. Radial error simulation results comparison between full FDM solving method and transfer function method (influence coefficient method). And the right hand figure shows those cosine and sine harmonic component amplitudes used for transfer function method.

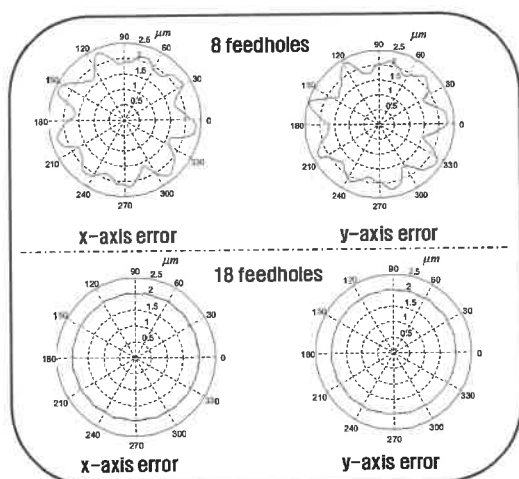
distribution variation and therefore bearing shaft error motion.

Figure 4 shows the schematic explanation of transfer function method which uses only a part of bearing segment for simulation and therefore faster and efficient calculation can be possible. Figure 5 shows the error motion of the journal shaft in the x and y direction due to the form error on journal surface and those two simulating methods are compared.

In order to investigate air bearing table's performance according to the number of feedholes (around the journal surface), a measured journal surface profile was used for analysis. In Fig. 6(a) the surface profile data and its spatial frequency components are shown for utilizing the transfer function method. As seen in the figure, higher components than tenth harmonic frequency have relatively lower magnitude of surface profile error. By both the magnitude of transfer function of a frequency and the magnitude of profile error amount, as seen in the Fig. 2(b), with eight feedholes the rotary table has larger error motion than in the case of eighteen feedholes. And also the error motion certainly has dominant frequency profile which can be explained by the transfer function analysis. With eighteen feedholes the rotary table's error motion amount is much decreased compared with the eight feedhole case.



(a)



(b)

FIGURE 6. (a) showing the measured surface profile error of journal shaft and its spatial frequency component's magnitudes. (b) with the measured surface profile the error motion of rotary table is calculated with the transfer function method.

TEST RIG FOR EXPERIMENT

Experiments will be done for stiffness, damping and run-out error assessment and a few sets of bearing and shaft types are available for varying the feedhole configuration, shaft roundness and so on.



FIGURE 7. The test rig for experimental investigation and assessment

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Three-Dimensional Surface Profile Measurement by Shape-from-Focus Method Using MEMS Mirror Projector

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INTRODUCTION

Noncontact three-dimensional surface profile measurement is used for shape inspections of industrial components. Most of the measurement method is based on a triangulation method such as stereo matching, light cut, and pattern-projection method. Recently, it is necessary to keep a space of production lines, reducing the size of the instruments. However, the triangulation method is difficult to reduce the size of instruments because it requires two optical axes such as a projection axis and an imaging axis. In order to overcome the problem, the Shape-from-Focus method (SFF) which is a uniaxial measurement method has been proposed. [1]. The SFF is a measurement method utilizing defocus of an imaging lens. The defocus depends on a distance between a focal point and an object surface. Therefore, the distance can be obtained by quantifying the defocus which defined as a contrast of a captured image. In order to detect the contrast, a grating pattern is projected onto an object surface. However, there is a problem to reduce the size of instruments because a large space is required due to the grating and the projection lens. In this paper, to overcome the problem, a micro-electromechanical system (MEMS) mirror is utilized to project the grating pattern instead of the grating and the projection lens for reducing the size of instruments. The MEMS mirror is a tiny silicon device for scanning light. The mirror is connected to a small transducer allowing it to oscillate. The MEMS mirror oscillates vertically and horizontally to project an image pixel-by-pixel. Therefore, the MEMS mirror projector is suitable for reducing the size of the instruments. In addition, an grating pattern can be projected due to shifting the projection images by the MEMS mirror and a personal computer. The purpose of this study is to develop a compact-size instrument to measure three-dimensional surface profile by SFF using the MEMS mirror projector.

PRINCIPLE OF THE SFF USING MEMS MIRROR PROJECTOR

Figure 1 shows the principle of the surface profile measurement by SFF. Figure 1(a) shows the formation of focused and defocused images. The object surface C at z_C is focused by the imaging lens which focal length is f . The surfaces A and B at z_A and z_B are defocused. Defocus value can be defined by using contrast value. The distribution of the contrast V with the distance z is represented as Bessel function as shown in fig. 1(b). The contrast V increases with approaching a focal point, and it peaks at a focal point $z = z_C$. And the contrast V decreases as the distance from a focal point increases. Thus, the contrast V_A , V_B and V_C correspond to a distance z_A , z_B and z_C . The surface profile is measured by quantifying the defocus as the contrast of the texture on the object surface by image processing.

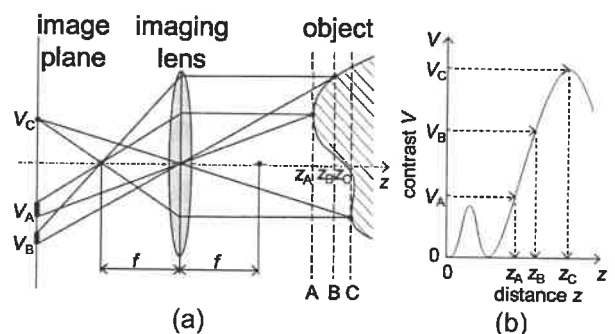


Figure 1. Principle of surface profile measurement by SFF. (a) Formation of focused and defocused images; (b) relationship between the contrast V and distance z .

Figure 2 shows the principle of the surface profile measurement by SFF using MEMS mirror projector. The measurement system consists of the MEMS mirror projector which is assembled from a laser and the MEMS mirror, an imaging lens, a camera, and a beam splitter. In order to calculate the contrast, a light intensity of projected pattern is needed to be sinusoidally distributed. Its distribution is generated by a

grating pattern, and the laser light is scanned and synchronized with on-off control of the laser. And then the pattern is projected onto a measuring object surface, and a sample image is captured by the camera through the beam splitter. It is associated with focal distance of an imaging lens. The defocus is quantified as the contrast by analyzing a projected sinusoidal image.

In order to measure the contrast, 4-step phase-shifting technique is employed. The sinusoidal intensity of light $I_\phi(x, y)$ projected onto the surface is given by

$$I_\phi(x, y) = A(x, y) + B(x, y) \sin(2\pi sy + \phi), \quad (1)$$

where $A(x, y)$ is DC component, $B(x, y)$ is an amplitude, s is a frequency of the sinusoidal pattern, and ϕ is a phase of the sinusoidal pattern. The contrast $V(x, y)$ is given by

$$V(x, y) = \frac{2\sqrt{(I_{0^\circ} - I_{180^\circ})^2 + (I_{90^\circ} - I_{270^\circ})^2}}{I_{0^\circ} + I_{90^\circ} + I_{180^\circ} + I_{270^\circ}}, \quad (2)$$

where I_{0° , I_{90° , I_{180° and I_{270° are light intensities of the sinusoidal pattern which phases are 0, 90, 180, and 270 degrees, respectively.

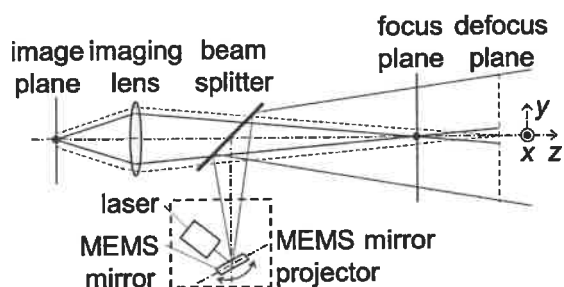


FIGURE 2. Principle of surface profile measurement by SFF using MEMS mirror projector.

EXPERIMENTAL SETUP

We have developed the three-dimensional surface measurement system by SFF utilizing the MEMS mirror projector. Figure 3 shows an optical setup of the instrument. It consists of the projection section and the imaging section. The projected sinusoidal pattern is generated by the MEMS mirror projector and its pattern can be controlled by a personal computer. The MEMS mirror projector is 50 mm in width and 90 mm in length. It has a resolution of 640×480 pixels. The light source is a 635 nm wavelength laser diode.

The size of the packaged MEMS mirror is 10×10 mm² and the mirror size is 1.2 mm in diameter. First, the MEMS mirror scans an object surface on the x-axis at the resonance frequency of 16500 Hz, the scanning angle of $\pm 10^\circ$. Then it scans on the y-axis at the resonance frequency of 1700 Hz, the scanning angle of $\pm 10^\circ$. The sinusoidal pattern is reflected by the beam splitter (THORLABS INC., CM1-BP145B1), and then it is projected onto an object surface. The object surface is imaged by imaging lenses which consist of two lenses and the CMOS monochrome camera (The Imaging Source Europe GmbH, DMK21BUC03). Its resolution is 640×480 pixels, with a maximum frame rate of 60 frames/s, and the data depth of 8 bits. The imaging lenses are 25.4 mm diameter plano-convex lenses. The focal length of lens F_1 is 50 mm, and that of lens F_2 is 750 mm. The distance between lenses is 46 mm. The dimensions of the instrument are 160 mm in height by 190 mm in length and, 139 mm in width.

In this paper, to generate an ideal sinusoidal pattern, a binary grating pattern is projected. The binary pattern can be regarded as a square wave horizontally onto the object surface plane, and an imaging system can be regarded as a point-spread function (PSF). The PSF can be approximated as Gaussian smoothing filter. It has been proposed by Lei *et al.* that by applying Gaussian smoothing filter to the square wave, the sinusoidal waveform can be generated [2]. We detected the binary pattern as a defocused pattern by a resolution limit of the imaging system. The defocused pattern is regarded as a square wave and a PSF of a convolution. Thus, it is the same as the ideal sinusoidal pattern.

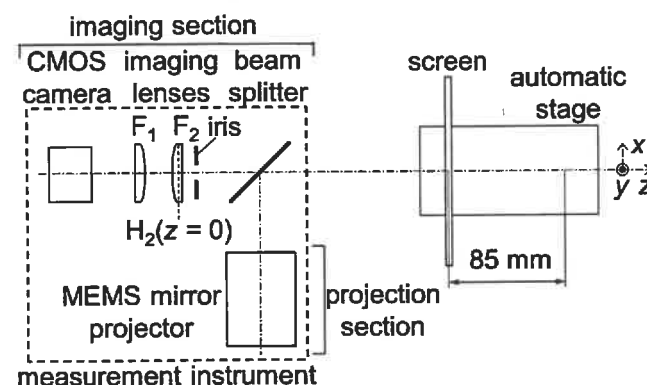


FIGURE 3. Optical setup by SFF utilizing the MEMS mirror projector.

In order to investigate the distance between a focal point and an object surface varied with the

contrast, the reference is obtained by moving a screen and an automatic stage. A screen is set on the z-axis of the stage, and then the binary pattern is projected onto the screen. The contrast is measured by 4-step phase-shifting technique. The z-axis of the automatic stage is driven by a stepping motor. The position of the automatic stage ranged from 0 to 85 mm, and it is controlled with a resolution 1 μm on the z-axis. The contrast V is measured at each position on the z-axis.

MEASUREMENT RESULTS

Figure 4 shows images of projected binary pattern on a screen in 4-steps phase-shifting technique. For detecting the defocused binary pattern, the contrast was calculated by 4-step phase-shifting technique.

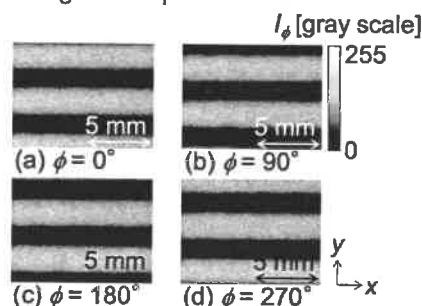


Figure 4. Detected image of projected binary pattern in 4 steps by a CMOS camera.

The contrast V is closely related the resolution of an imaging system which is determined by numerical aperture (NA). Thus, the relationship between the contrast and NA has been measured. Figure 5 shows the z-axial contrast distribution for the case of NA = 0.05 and 0.16 lenses by changing the diameter of the iris shown in fig.3. The contrast on the screen was measured by 4-step phase-shifting technique. The pitch of projected binary pattern is 0.288 lines/mm. The principal point H_2 of the imaging lens F_2 set $z = 0$ as shown in fig.3. The automatic stage was moved in the range from $z = 80$ to 164 mm by 1.0 mm. The contrast plotted in fig. 5 is the average in a detected 2D image. The variation of the contrast V with the distance z is represented as Bessel function. We define the range in which the contrast monotonically increases in lying to the left of peak as the z-axial measurement range. Comparing the distribution of NA = 0.05 with that of NA = 0.16, the variation of the contrast with the variation of the distance is larger in NA = 0.16. The contrast distribution of using NA = 0.16 is substantially increases. Thus, the z-axial resolution of NA =

0.16 is higher than that of NA = 0.05. However, the defined measurement range of NA = 0.16 is shorter than that of NA = 0.05. The each measurement range of NA = 0.16 and 0.05 are 26 mm and 50 mm, respectively. In this paper, to obtain the measurement range as large as possible, NA = 0.05 lenses is set up in measurements.

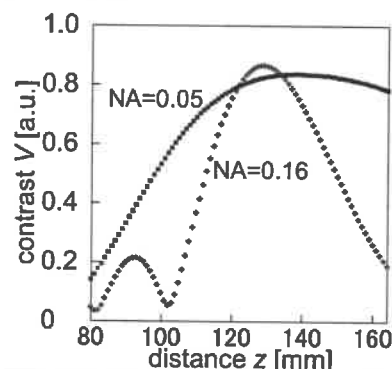


Figure 5. Contrast distribution for the case of projecting the binary pattern in difference NA.

In order to investigate the pitch of the binary pattern which is a suitable for increase of the measurement range, the contrast has been measured with changing the pitch. Figure 6 shows the variation of the z-axial contrast distribution changing the pitch. The pitch was changed from 0.288 to 0.683 lines/mm. The variation of the contrast with z position projecting a grating pattern in the pitch of 0.288 lines/mm was the highest, and it was seen that the contrast had peaks at $z = 140$ mm. As the decreasing the pitch of the binary pattern, a peak of contrast became lower. Thus, the pitch of 0.288 lines/mm was a suitable for increase of the measurement range. A distance z can be determined by obtaining the relation between the contrast and the distance by the nonlinear fitting based on Levenberg-Marquardt algorithm. The range which the contrast monotonically increases in can be fitted. As a result, the measurement range is a 60.0 mm range by fitting the contrast distribution of the pitch of 0.288 lines/mm. And then we checked an accuracy of the instrument by applying the fitting.

Figure 7 shows an accuracy check of the instrument. The measurement value was obtained by the nonlinear fitting. The value plotted in fig.7 is representative value at $(x,y) = (200,200)$ in detected 2D image. The sinusoidal grating pattern of 0.288 lines/mm was projected on the screen had set in the automatic stage, and then the automatic motor stage was moved

in the range from $z = 80$ to 164 mm by 1.0 mm. The solid line shown in fig.7 is an ideal line. The accuracy was 0.5 mm at $(x,y) = (200,200)$. Measuring at all (x,y) positions, the accuracy was ± 2.5 mm.

For the evaluation of the system, the surface profile of the sample was measured. Figure 8 shows the groove-cutting sample with a depth of 25.0 mm. The sample consisted of the diffuse reflective surface which manufactured of aluminum materials, and sprayed with a lacquer coating of matte white. Figure 9 shows a surface profile of the sample. The size of the measured area was 80×43 mm². Figure 9(a) shows the contrast distribution of the sample. The contrast is difference between the top flat and the bottom flat. Figure 9(b) shows a surface profile of the sample. The measured depth is 25.0 mm. However an accuracy in the bottom flat is ± 10.0 mm. Because the contrast V is high in the bottom flat, and a small changing the contrast V leads to a significant change in the obtained distance z in the high contrast V toward the peak as shown in fig. 6.

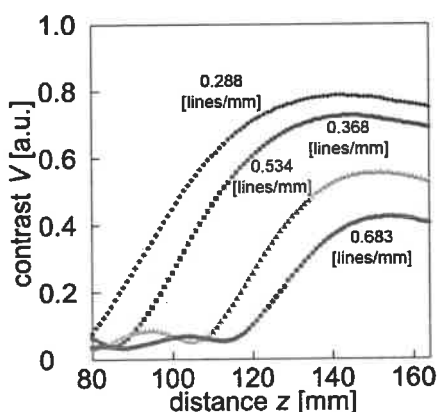


FIGURE 6. Contrast distribution changing a pitch of a binary pattern.

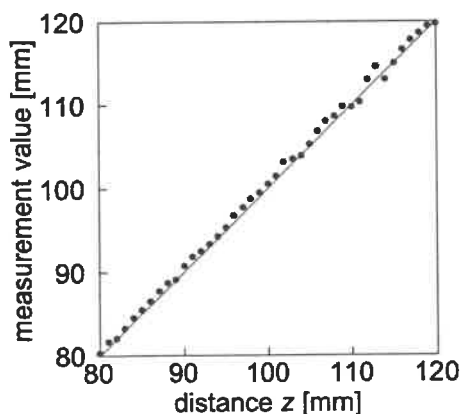


FIGURE 7. Accuracy of the instrument.

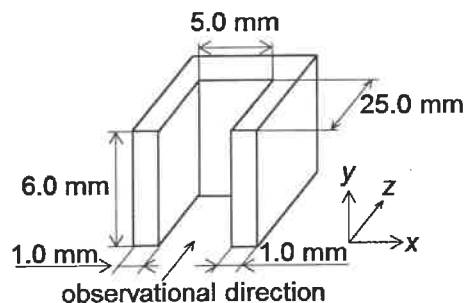


FIGURE 8. Groove-cutting sample.

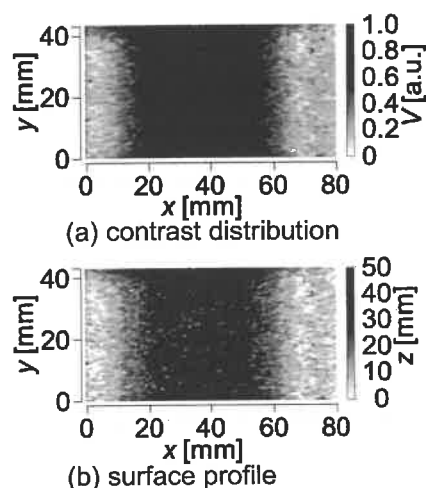


FIGURE 9. The surface profile of the sample. (a) The contrast distribution of the sample; (b) the surface profile of the sample.

CONCLUSION

The three-dimensional surface measurement instrument by SFF using the MEMS mirror has been developed. Measuring the contrast value z axially changes, the relation between the contrast value and the distance were obtained. The accuracy of the instrument was ± 2.5 mm. For the evaluation of the system, the surface profile of the groove-cutting sample can be measured. It was successful to measure its depth of 25.0 mm. However an accuracy lowered in the bottom flat. The problem is presumably due to the fact that the variation of the distance z is strongly affected by that of contrast V which is a high value.

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MEASUREMENT AND COMPENSATION OF BAR-MIRROR FLATNESS AND SQUARENESS ERROR USING HIGH PRECISION STAGE

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INTRODUCTION

In precision stage systems, laser interferometers are used to measure positions of the stage. Laser interferometers have to be used with bar-mirrors to reflect laser. As the length of bar-mirrors is lengthen, the flatness of bar-mirror become large.

Setup and environment of experiment such as temperature variation are also the reason of errors. The deformation of bar-mirror's shape cause straightness error, and setup error cause squareness error of precision stages. So, to improve precision stage's performance, error measurement and compensation are necessariness.

To measure flatness profile of bar-mirror, the straightness motion of the stage must be considered. Because of the accuracy of the flatness profile of bar-mirror is at the same level as that of the straightness motion of the stage. So, it is necessary to separate these two kinds of errors.

The flatness errors can be compensated by using measured data of bar-mirror's profile in real time.

The squareness error can be obtained using the polygon mirror.

In this paper, measure the flatness error of bar-mirror and the squareness error of precision stage. Compensate those and evaluate errors of precision stage.

PRINCIPLES OF FLATNESS AND SQUARENESS ERROR

Flatness Error Measurement

There are many methods to separate these errors and to measure workpiece flatness profiles. The inclination method [1] and the generalized two-point method [2] have been proposed for that purpose. In these methods, two displacement probes are used to measure

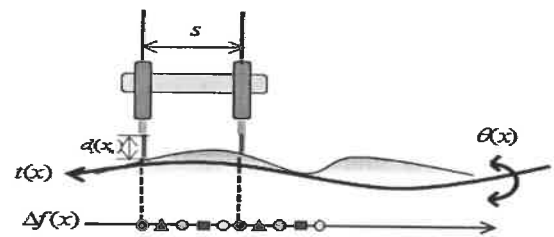


FIGURE 1. Principle of the generalized two-point method

the flatness profile. The straightness motion of the stage is canceled by the differences of probes' output, and the flatness profile of bar-mirror is obtained by simple data processing operations. The combined method [3], which combines the advantages of the inclination method and the generalized two-point method, is proposed. Also, to realize high accuracy profile measurement many kinds of three-point methods [4, 5] are proposed.

The generalized two-point method is selected to measure the flatness profile of bar-mirrors. Figure 1 shows the principle of the generalized two-point method schematically. Two displacement sensing probes are fixed and can measure the flatness profile of bar-mirror while the stage moves. Assume that the flatness profile of bar-mirror is $f(x)$, the straightness motion of the stage is $t(x)$, and the yaw motion of the stage is $\theta(x)$. If the output of probe 1 at point x_n is denoted by $d_1(x_n)$, and output of probe 2 at point $x_n + s$ is denoted by $d_2(x_n)$, $d_1(x_n)$ and $d_2(x_n)$ are denoted as

$$\begin{aligned} d_1(x_n) &= f(x_n) + t(x_n) \\ d_2(x_n) &= f(x_n + s) + t(x_n + s) + s \times \theta(x_n) \end{aligned} \quad (1)$$

Find the remainder of $d_1(x_n)$ and $d_2(x_n)$, $t(x_n)$ is canceled.

$$\begin{aligned} F(x_n) &= f(x_n + s) - f(x_n) \\ &= d_2(x_n) - d_1(x_n) - s \times \theta(x_n) \end{aligned} \quad (2)$$

A derivative of d can be defined as follows

$$\begin{aligned} F'(x_n) &= F(x_n) / s \\ &= (d_2(x_n) - d_1(x_n) - s \times \theta(x_n)) / s \end{aligned} \quad (3)$$

The integration of $F'(x_n)$ represents the profile of bar-mirrors.

$$\begin{aligned} p(x_n) &= \sum_{i=1}^n F'(x_i) s \\ &= p(x_{n-1}) + F'(x_n) s \\ &= p(x_{n-1}) + (d_2(x_n) - d_1(x_n) - s \times \theta(x_n)) s \end{aligned} \quad (4)$$

Where $d_1(x_n)$, $d_2(x_n)$ and s are known, $\theta(x_n)$ can be measured by angular sensor.

Squareness Error Measurement

To measure squareness error, the reference polygon mirror is needed. Fig 2 shows experiment setup and the process to measure squareness error. The polygon mirror is fixed on moving part of stage, and measured by displacement sensor during move X and Y axis direction. After 90-degree rotate the polygon mirror on Z axis, measure the polygon mirror's surface. Squareness error α can be obtained by total four times measurement and equation 6.

$$\begin{aligned} \alpha &= \beta_1 + \theta_{x1} + \theta_{y1} \\ \alpha &= \beta_2 + \theta_{x2} + \theta_{y2} \\ \alpha &= \beta_3 + \theta_{x3} + \theta_{y3} \\ \alpha &= \beta_4 + \theta_{x4} + \theta_{y4} \end{aligned} \quad (5)$$

$$\alpha = 90^\circ + \frac{1}{4} \left(\sum_{i=1}^4 \theta_{xi} + \sum_{i=1}^4 \theta_{yi} \right) \quad (6)$$

θ is the angle between polygon mirror's surface and bar-mirror's surface. β is the inside angle of polygon mirror. So the sum total of β is 360-degree.

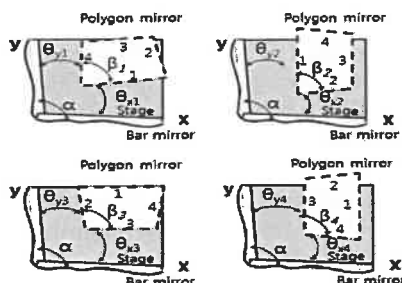


FIGURE 2. Schematic diagram of the experiment to measure the squareness error

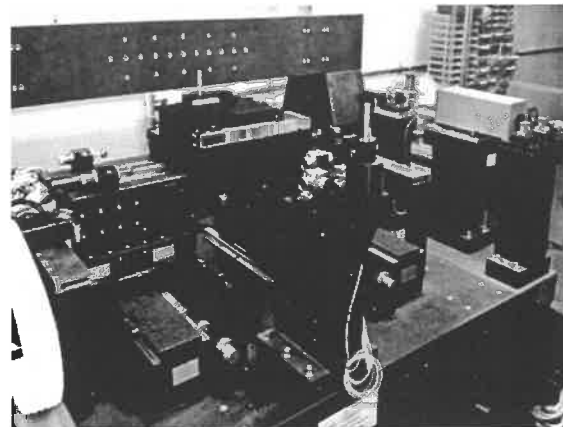


FIGURE 3. Photograph of the precision stage used to experiment

EXPERIMENT

Experimental setup

The experiment system consists of the following parts. Two laser type displacement sensors to measure polygon mirror's surface are used. Those measure the surface of polygon mirror that is fixed on the precision stage.

Two bar-mirrors are fixed X, Y precision stage. The precision stage is actuated by three linear motors. One linear motor actuates the stage in X direction, and the others actuate in Y direction. Three laser interferometers are used to measure the position of precision stage. One measures the position of stage in X direction, and the others measure in Y direction and θ_z .

The stage's moving range is 200mm x 250mm.

Flatness Measurement Experiment

Each bar-mirror's length is 300 mm and moving range of stage is 200 mm. The displacement sensor interval is 35 mm and sampling period is 5 mm.

Figure shows the reconstructed profile of two bar-mirrors using generalized two-point method.

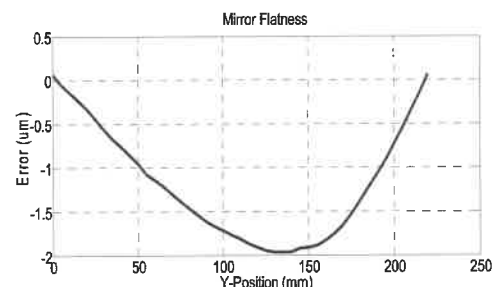


FIGURE 4. Reconstructed profile of bar-mirror's flatness error

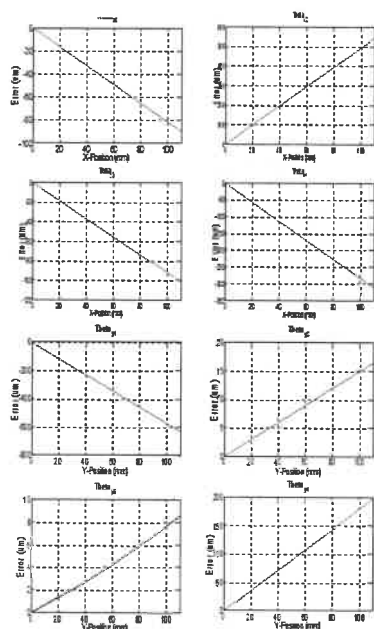


FIGURE 5. Sampling data of squareness error measurement experiment

Squareness Measurement Experiment

The size of polygon mirror is 150mm x 150mm, and the sampling range is 110mm x 110mm. The sampling period is 10 mm.

The slope of sampling data can be obtained using least square line.

Figure 5 shows sampling data and the slope is arranged in table 1. Using eq. 6 and the data of table 1, squareness error is -0.192 degree.

TABLE 1. The slope of squareness error measurement experiment's data

θ_{x1}	-0.047	θ_{y1}	-0.330
θ_{x2}	+0.284	θ_{y2}	+0.009
θ_{x3}	-0.053	θ_{y3}	+0.004
θ_{x4}	-0.162	θ_{y4}	+0.103

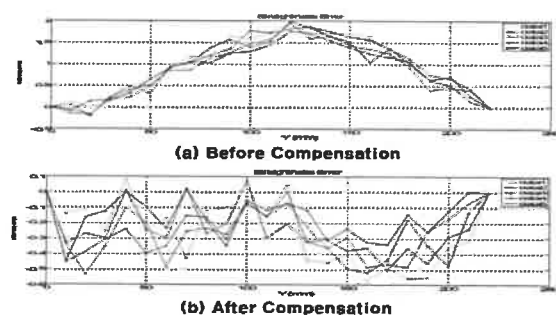


FIGURE 6. The straightness error of the precision stage

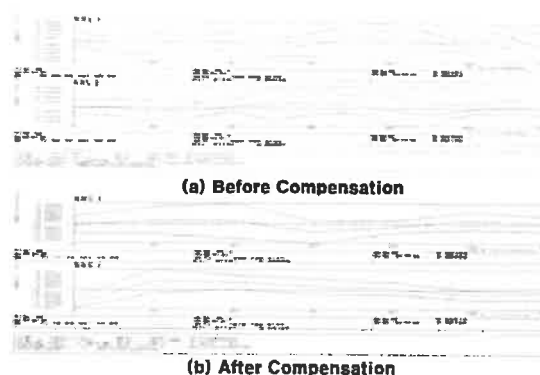


FIGURE 7. The squareness error of the precision stage

Evaluation

To evaluate experiment result, laser calibrator (Renishaw, ML10) is used.

Straightness and squareness errors are measured five times each. Figure 6 is the result of straightness error measurement and figure 7 is the squareness error.

In table 2, there are quantities of errors before and after compensation.

TABLE 2. The quantities of errors before and after compensation

	Before	After
Straightness Error	2.71 μm	741 nm
Squareness Error	169.4 arc-sec	11.2 arc-sec

Conclusion

In this paper, performances of precision stage are improved by error compensation. To reduce straightness and squareness error, bar-mirror's flatness and squareness error are measured and compensated. And after evaluate precision stage's straightness error using laser calibrator.

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